

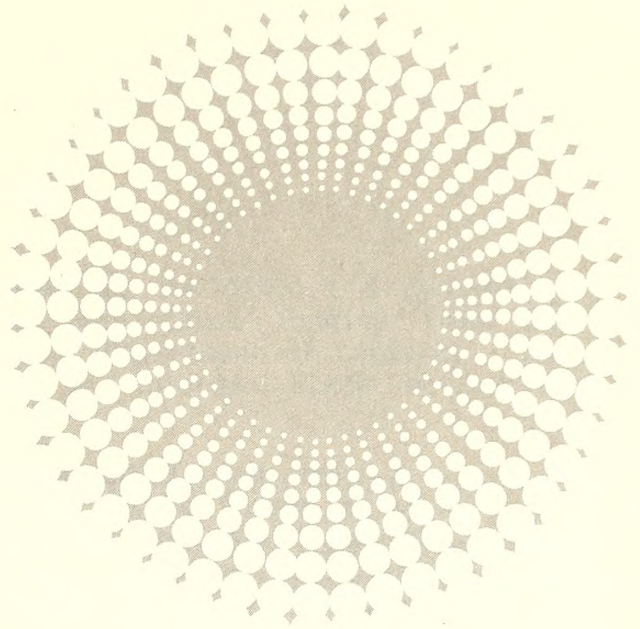
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SOLAR HEATING AND COOLING OF RESIDENTIAL BUILDINGS **DESIGN OF SYSTEMS**

1980 EDITION



U.S. DEPARTMENT OF COMMERCE
Economic Development Administration



SOLAR HEATING AND COOLING OF RESIDENTIAL BUILDINGS **DESIGN OF SYSTEMS**

1980 EDITION

Prepared by
SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY



U.S. Department of Commerce
Phillip M. Klutznick, Secretary
Luther H. Hodges, Jr., Deputy Secretary
Robert T. Hall, Assistant Secretary
for Economic Development

SEPTEMBER 1980

NOTICE

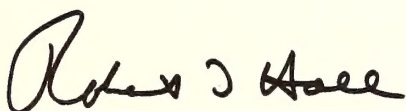
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FOREWORD


"Solar Heating and Cooling of Residential Buildings," a two-volume manual, was developed with technical assistance funding from the Economic Development Administration, U.S. Department of Commerce. The Solar Energy Applications Laboratory of Colorado State University prepared the materials.

Since its publication in 1977, the first edition has provided practical information for architects, mechanical engineers, contractors and technicians concerned with the steady rise of energy costs. Wide acceptance of the manual marks a contribution by EDA toward nationwide efforts to help conserve America's hard pressed energy resources.

The present edition of both volumes -- "Design of Systems" and "Sizing, Installation and Operation of Systems" -- updates the first edition. New material reflects recent research and practical experience in the fast-growing field of solar energy -- a form of energy that can contribute substantially to the reduction of fuel imports, lead to increased production of housing, and create opportunities for expanded employment.

A handwritten signature in dark ink, appearing to read "Robert T. Hall". The signature is fluid and cursive, with the first name "Robert" being more prominent than the last name "Hall".

Robert T. Hall
Assistant Secretary
for Economic Development



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PREFACE

This manual was prepared primarily for use in conducting a practical training course on the design of solar heating and cooling systems for residential and small office buildings, but may also be useful as a general reference text. The content level is appropriate for persons with different and varied backgrounds, although it is assumed that readers possess a basic understanding of heating, ventilating, and air-conditioning systems of conventional (non-solar) types.

This edition is a revision of the manual with the same title, first printed and distributed by the U. S. Government Printing Office in October 1977. The manual has been reorganized, new material has been added, and outdated information has been deleted.

If this manual is used for a practical level training course, there should be need for a minimum of supplementary material to conduct the course. The content level has been carefully evaluated and tested to be appropriate for practitioners in the building industry, such as contractors, engineers, and architects.

Only active solar systems are described in this manual. Other text-books and workshop manuals are available for passive designs. Liquid and air-heating solar systems for combined space and service water heating or service water heating only are included in this manual. Furthermore, only systems with proven experience are discussed to any extent.

The training course curriculum and this manual for Design of Systems for space and water heating were developed by the staff of the Solar Energy Applications Laboratory and vocational education

specialists at Colorado State University in cooperation with the NAHB Research Foundation. A national advisory committee selected from various sectors of the home-building industry, university sources, private practice, and government, was established to provide advice and general guidance to the CSU project staff.

ACKNOWLEDGMENTS

The original manual was prepared by the following staff members of the Solar Energy Applications Laboratory at Colorado State University: C. Byron Winn, Susumu Karaki, George O. G. Löf, Gearold Johnson, Dan S. Ward, William S. Duff, and Sanford B. Thayer. Ivan E. Valentine and Milton E. Larson are the vocational education specialists at Colorado State University who participated in the program, and Ralph J. Johnson and H. W. Anderson of the NAHB Research Foundation provided expert advice regarding home building practices. The present edition was revised and rewritten by Susumu Karaki and George Löf.

Development of the training course and preparation of the original manual were made possible by a matching grant from the Economic Development Administration, U.S. Department of Commerce, with Colorado State University providing the matching funds. Financial assistance was also received from the U.S. Department of Housing and Urban Development for video-taping lectures during early trial presentations.

Financial assistance for preparing this revised manual was received from the Solar Heating and Cooling Research and Development Branch, Office of Conservation and Solar Applications, U. S. Department of Energy. Vincent Rice was the project monitor.

Special appreciation is expressed to Diana Rose and Wendy Asa for their diligence and perseverance in typing and preparing the manuscript. Their patience and service are truly appreciated.

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TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 1

COURSE OUTLINE

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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OBJECTIVES

The training course is designed to develop the capability of each trainee to design solar heating and cooling systems for small buildings used as residences and offices. Specific objectives of the course are to develop capabilities of trainees to:

1. Identify different types of solar systems and their components.
2. Differentiate between experimental and proven systems.
3. Design solar systems for heating space and service water.
4. Calculate expected system performance of a specified solar heating system.
5. Recognize installation procedures and problems.
6. Measure and evaluate system performance.
7. Recognize maintenance requirements.
8. Explain system operation to system owners.

INTRODUCTION

The Solar Energy Applications Laboratory at Colorado State University prepared this manual to be used in conjunction with a training course on the practical aspects of designing solar systems for space and service water heating. In determining curriculum content, a rigorous procedure was followed to (1) develop course standards, (2) establish needs of contractors, engineers and architects to design and install solar systems, (3) delineate objectives for the course and (4) develop the curricular materials. The manual is modularized for

flexibility in conducting the course and it is not essential (although preferable) to follow the modules in numerical order.

So that the order of the modules can be rearranged, and so that the material may be conveniently used as a reference manual, there is some repetition and duplication. For example, components of liquid and air systems are covered not only in Modules 7 and 8 but also in the sections on heating systems (Modules 2 and 10). When the manual is used as a course textbook, these duplications may be employed as review material or they may be passed over.

At Colorado State University the training course is conducted in five consecutive days, but the period of presentation may be varied to suit the needs at a particular location. Evening hours may be utilized during which one or more modules can be presented and discussed, or a shorter period than five days may be chosen and some material may be deleted. Each module has stated objectives, and examinations may be given to determine achievement levels of the trainees.

COURSE SCHEDULE

A course arrangement suitable for a five-day period is suggested on the following page. The modules are arranged, first to introduce trainees to fundamental concepts of solar systems and components, and then provide an approximate method of sizing systems. This is followed by a more detailed discussion of components and a detailed method of system design. After methods to estimate performance are discussed, economic issues are addressed. Installation concerns and system performance measurement procedures conclude the course.

T:time	SUNDAY	MONDAY	TUESDAY	WEDNESDAY	THURSDAY	FRIDAY
0830		MODULE 1 (30 min) Course Outline ----- MODULE 2 (60 min) Introduction to Solar Heating Systems	QUESTIONS & ANSWERS (30 min) ----- MODULE 6 (60 min) System Sizing and Approximate Methods	QUESTIONS & ANSWERS (30 min) ----- MODULE 9 (60 min) Domestic Hot Water Systems	MODULE 15 (60 min) Operational Checkout ----- Laboratory (150 min)	MODULE 14 (90 min) Installation Concerns
1000		BREAK (30 min)	BREAK (30 min)	BREAK (30 min)		BREAK (30 min)
1030		MODULE 3 (90 min) Solar Radiation	MODULE 6 (continued) (90 min)	MODULE 10 (90 min) Solar Heating Systems		MODULE 16 (60 min) Fundamentals of Solar Cooling ----- COURSE EVALUATION (30 min)
1200		LUNCH (60 min)	LUNCH (60 min)	LUNCH (60 min)	LUNCH (60 min)	ADJOURN
1300		MODULE 4 (120 min) Solar Collectors	MODULE 7 (120 min) Components of Liquid Systems	MODULE 11 (120 min) Detailed Design Method	MODULE 12 (120 min) Economic Analysis	
1330	Registration					
1400	Solar House Tour (180 min)					
1500		BREAK (30 min)	BREAK (30 min)	BREAK (30 min)	BREAK (30 min)	
1530		MODULE 5 (90 min) Heating and Cooling Loads	MODULE 8 (90 min) Components of Air Systems	MODULE 11 (continued) (90 min)	MODULE 13 (90 min) Alternative Economic Analyses	
1700		ADJOURN	ADJOURN	ADJOURN	ADJOURN	
1730	Reception and Dinner				Dinner	
1900	Passive Solar Designs (45 min)				MODULE 17 (60 min) Future Prospects	

COURSE CONTENT

TOUR OF SOLAR HOUSES

A pre-course tour of solar houses is provided to give trainees an opportunity to see different types of systems and different styles of solar homes. The duration of the tour is about three hours.

MODULE 1 - COURSE OUTLINE

The objectives of the course are discussed, and course schedule is presented. The purpose is to outline clearly to participants what is expected in the course.

MODULE 2 - INTRODUCTION TO SOLAR HEATING AND COOLING SYSTEMS

This module presents the types of solar systems covered in this manual and training course. Because of the varied backgrounds expected of trainees, this introductory module is used to establish a common foundation for all trainees in the course.

MODULE 3 - SOLAR RADIATION

A procedure to calculate solar radiation on tilted collectors is the principal content of the module. It is also important to understand the variability of the energy source.

MODULE 4 - SOLAR COLLECTORS

The basic principles of flat-plate solar collectors are discussed in this module. Parameters which characterize collector performance are explained and factors which affect efficiency are discussed. Differences between air and liquid-type collectors are explained.

MODULE 5 - HEATING AND COOLING LOAD ANALYSES

For sizing furnaces, boilers, and air-conditioners for building space conditioning systems it is advisable to make heating and cooling load calculations. To design economical solar systems, such calculations are essential. Simple procedures for heating and cooling load analyses are described in the module.

MODULE 6 - SYSTEM SIZING AND APPROXIMATE METHODS FOR ESTIMATING SYSTEM PERFORMANCE

Collector areas are selected arbitrarily or based upon economic criteria. Component sizes are based on collector area and simple rules. These design rules are presented and approximate methods to estimate system performance are described.

MODULE 7 - COMPONENTS OF LIQUID SYSTEMS

Components for air and liquid systems are basically different although their functions are similar. Storage heat exchangers, controls and pumps for liquid systems are discussed in this module.

MODULE 8 - COMPONENTS OF AIR SYSTEMS

Thermal energy storage, controls and blowers for air systems are described in this module.

MODULE 9 - DOMESTIC HOT WATER SYSTEMS

Different types of service water heating systems are described, operating characteristics are explained and a simple sizing procedure is presented.

MODULE 10 - SOLAR SYSTEMS FOR SPACE HEATING

Detailed operating characteristics of solar heating systems are described in this module.

MODULE 11 - DESIGN PROCEDURES

The f-chart method for estimating system performance is the detailed design method that is generally used. The procedure is flexible in that components of different sizes may be included in the system, but application is limited to specified types of solar systems.

MODULE 12 - ECONOMIC CONSIDERATIONS

Procedures for calculating life-cycle costs of solar and conventional heating systems are presented. Various factors pertinent to the analysis are also explained.

MODULE 13 - OTHER ECONOMIC CONSIDERATIONS

Since life-cycle costing involves many assumptions, more simple approaches in determining economic viability of solar systems are explained in this module.

MODULE 14 - INSTALLATION OF SOLAR SYSTEMS

Designers of solar systems should be generally aware of preferred installation procedures and problems which may arise during installation. Through understanding of installation problems, designs can be made to minimize installation costs.

MODULE 15 - OPERATIONAL CHECK-OUT

All systems must be checked after installation to determine that the system operates properly. Items of concern and procedures for orderly inspection and testing are described.

MODULE 16 - FUNDAMENTALS OF SOLAR COOLING

Various solar cooling alternatives are described in this module. Absorption cooling systems, Rankine-cycle systems and desiccant cooling are described.

MODULE 17 - FUTURE PROSPECTS FOR SOLAR HEATING AND COOLING SYSTEMS

Components of systems that are under development with prospects for future use are discussed.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 2

INTRODUCTION TO SOLAR HEATING AND COOLING SYSTEMS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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OBJECTIVES

At the end of this module, the trainee should be able to:

1. Identify the principal characteristics of solar heating and cooling systems.
2. Describe the basic functions of key components of solar heating and cooling systems.
3. Recognize the advantages and limitations of different designs of components and systems.

INTRODUCTION

The purpose of this module is to provide a brief introduction to the types of solar heating and cooling systems that are available and in current use, and to explain the basic functions of the systems and their key components. Methods of solar heat collection, storage, and control are outlined, and the use of these components in complete systems with auxiliary heat supply is described.

SOLAR HEATING AND COOLING SYSTEMS

A solar heating and/or cooling system can be defined as any system which utilizes solar energy to heat and/or cool a building, although a distinction is usually made between "active" and "passive" types. In contrast with architectural features which permit direct solar entry into the building, active systems require special equipment for collecting solar energy in liquid or air, storing heat, and distributing heat

to the rooms. They can be integrated directly into conventional HVAC systems in buildings with provisions for controlled collection and distribution of solar and auxiliary heat. Heat may be collected in a liquid, usually water, or in a gas, always air, for transport to storage or use. Both types are important and extensively used.

LIQUID-HEATING SOLAR SYSTEMS

A BASIC SYSTEM

Figure 2-1 is a simplified schematic drawing of a liquid-heating solar system, representative of those widely available today. The key components are a solar collector, a tank for heat storage in hot water, an auxiliary furnace, and a system for delivering heat to the building. Many solar heating systems also include a heat exchanger for transferring heat from a non-freezing collector fluid to water storage.

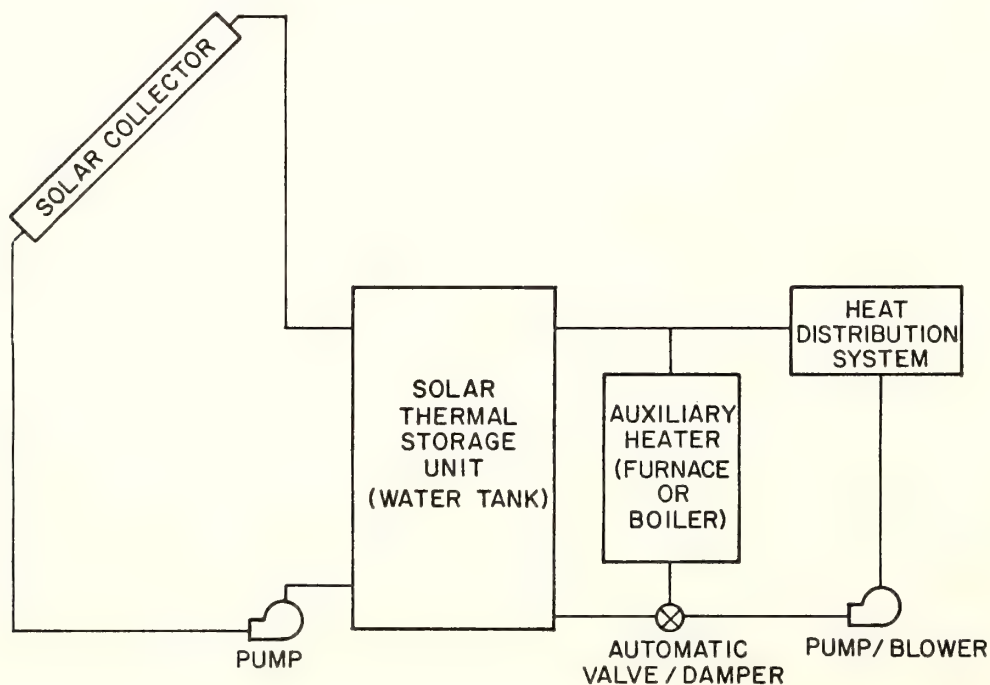


Figure 2-1. Schematic Diagram of a Simplified Liquid Type Solar Heating System

Facilities for providing solar heat to the domestic hot water supply are also commonly used.

Solar radiation is absorbed in the collector, the temperature of a liquid, usually water, being pumped through the collector increases, and the heated water is transferred to a storage tank (or, in some cases, directly to the heat distribution system). The thermal storage unit is usually essential because heat is ordinarily required at night and during cloudy periods. The solar collector and thermal storage unit can operate independently of heating demands, so solar energy can be collected and stored whenever there is sufficient incident solar radiation.

DRAIN-BACK SYSTEM

If water is used in the solar collectors in a cold climate, some freeze protection method must be provided. A common arrangement shown in Figure 2-2 permits the collector to drain into the storage tank whenever operation of the circulating pump stops. When the solar intensity is sufficient for heat collection, a pump circulates water through the collectors and the storage tank. When the pump shuts off, water in the collector drains into the storage tank while air enters the collector tubes through an atmospheric vent. An alternate design provides a down-flow pipe large enough for air from the top of the vented storage tank to rise through the pipe as water drains down when the pump operation ceases. In some systems, automatic valves operate to drain the collector only when freezing temperatures are approached.

DUAL-LIQUID SYSTEM

Figure 2-3 shows another method for freeze protection, by which solar heat is collected in a non-freezing liquid, usually a water-glycol

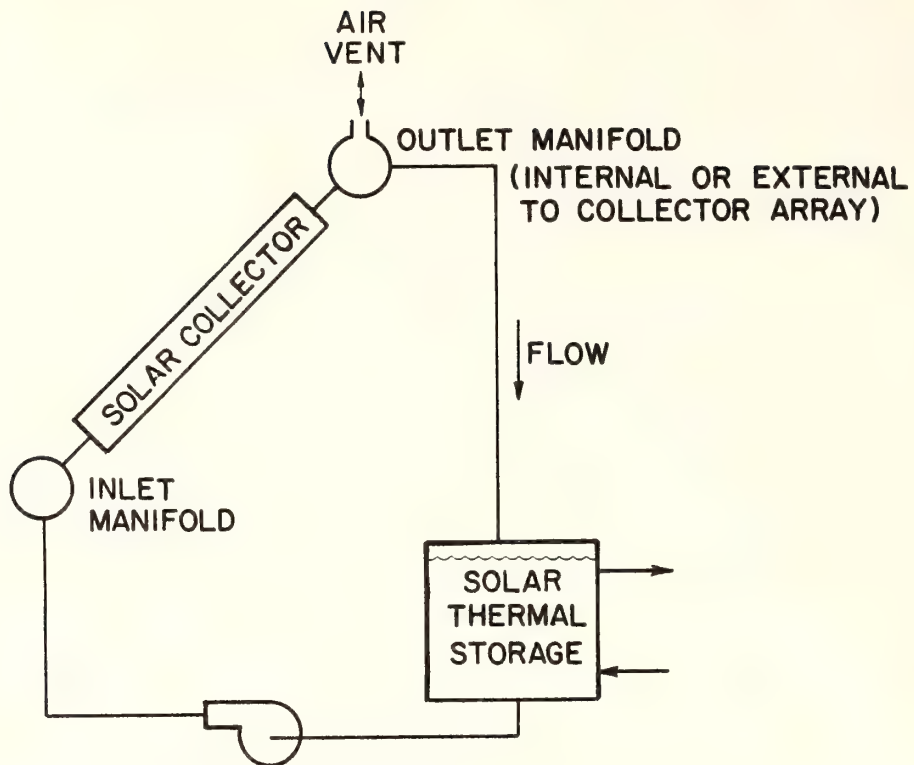


Figure 2-2. Drain-Down System for Solar Collection and Storage

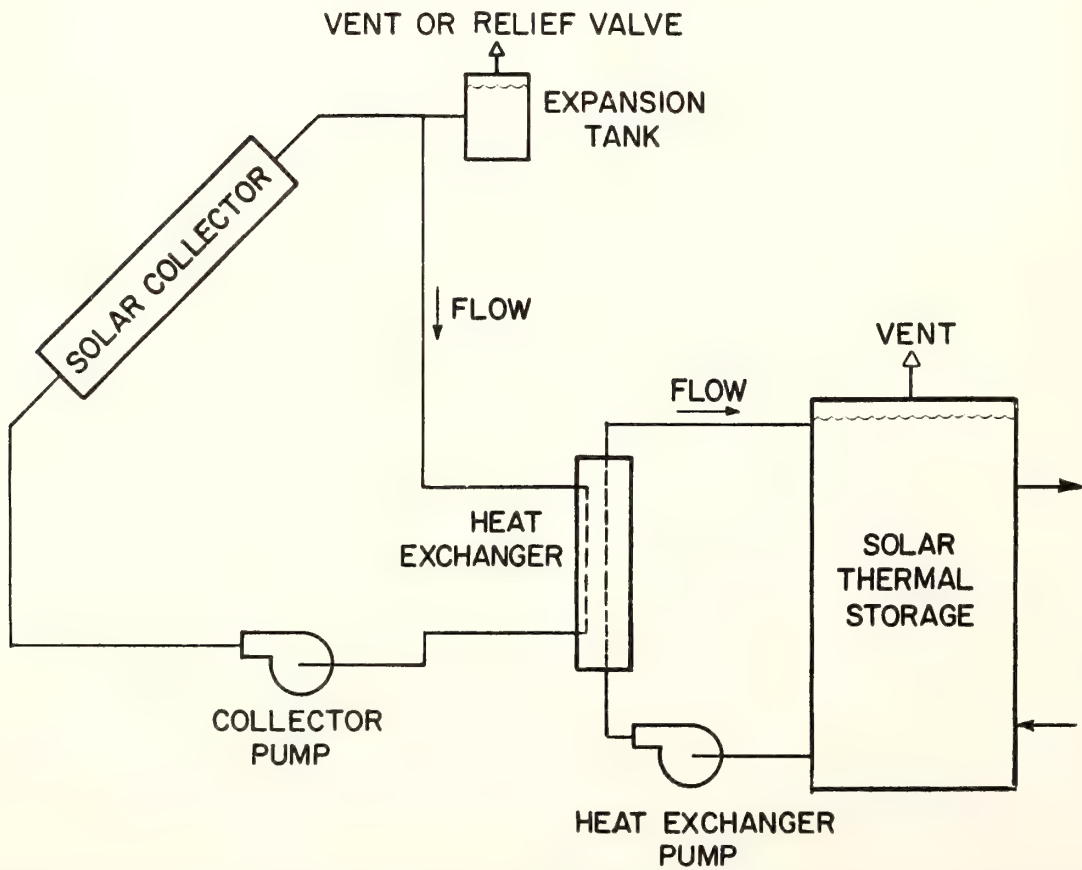


Figure 2-3. Solar Collection in Non-Freezing Liquid and Heat Exchange with Water Storage

solution. To avoid the cost of several hundred gallons of glycol anti-freeze in the storage liquid, water is used for heat storage and a heat exchanger is provided for transfer of heat from the collector fluid to storage. A second pump is required unless the heat exchanger is located inside the storage tank.

The advantages of this design are the minimal risk of freezing (and damage) from incomplete collector drainage or venting, and the absence of corrosion caused by alternate exposure of the collector tubes to water and air in the drain-back system. The possibility of corrosion and freezing in the drain-back system (in Figure 2-2) can thus be compared with the cost penalty of the heat exchanger, additional pump and piping, and temperature loss in the exchanger for the design in Figure 2-3.

NEED FOR AUXILIARY HEATING

An auxiliary furnace is required as a back-up to the heating system for use when the solar collector/thermal storage subsystem is unable to meet heating demands. Although the collector and storage could be sized large enough to meet the full heating requirements throughout the year, the cost would be prohibitive in most climates. An economical design involves a smaller solar system and an auxiliary furnace or boiler capable of meeting the full heating demand (at design conditions) when solar collection is not possible and when stored heat is inadequate. This combination results in more effective use of the solar equipment throughout the heating season and a lower annual total cost.

METHODS OF HEAT DELIVERY

Heat can be delivered to the building in several ways. With a warm-air distribution system, heat is usually supplied to air from stored hot water by use of a liquid-to-air heat exchanger ("fan-coil") with pump and blower for circulating the two fluids. Space heating may also be provided by circulating the hot water to all parts of the building where individual fan-coil units or "radiator" surfaces such as baseboard strip heaters (finned tubing) are used.

SERVICE WATER HEATING

Service hot water may be solar heated either in a system for that purpose only, or in combination with a space heating system. Figure 2-4

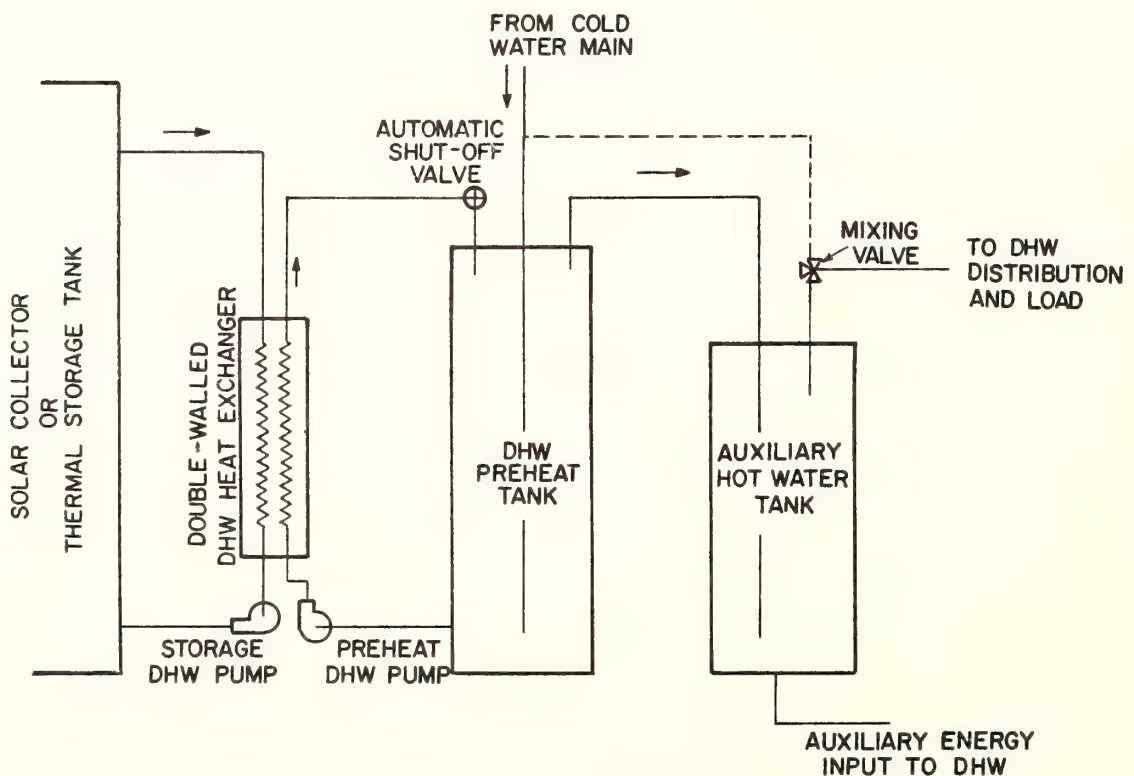


Figure 2-4. Schematic Arrangement of a Solar Domestic Hot Water (DHW) Subsystem

is a schematic drawing showing a typical arrangement of components for utilizing solar heat in a domestic hot water (DHW) system of either type. Solar energy from a collector or a thermal storage unit is used to preheat water supplied from the cold water main. Thermostatic controls actuate the two pumps when the DHW preheat tank is colder than the solar source so that heat can be transferred in the heat exchanger to the DHW supply. Instead of the separate heat exchanger, coils of tubing inside the DHW preheat tank may be used. A conventional hot water heater supplements the solar supply as required. As hot water is used in the building, water preheated by solar energy replaces the quantity drawn from the auxiliary hot water tank. Gas or electricity is used to raise and maintain the temperature of the preheated water at the desired setting (e.g., 140°F).

LIQUID SYSTEM COMPONENTS

In addition to the components of conventional heating systems, a solar system requires a solar collector, thermal storage unit, DHW preheat tank, some additional plumbing/sheet metal work, and a more extensive control system. Of the additional solar components, the most important is the solar collector.

Solar Collectors for Liquid Heating

A solar collector is a device to convert incident solar radiation to useful energy, usually in the form of heated air or heated liquid. Figures 2-5 and 2-6 show examples of two liquid-type solar collectors used in solar heating and cooling systems.

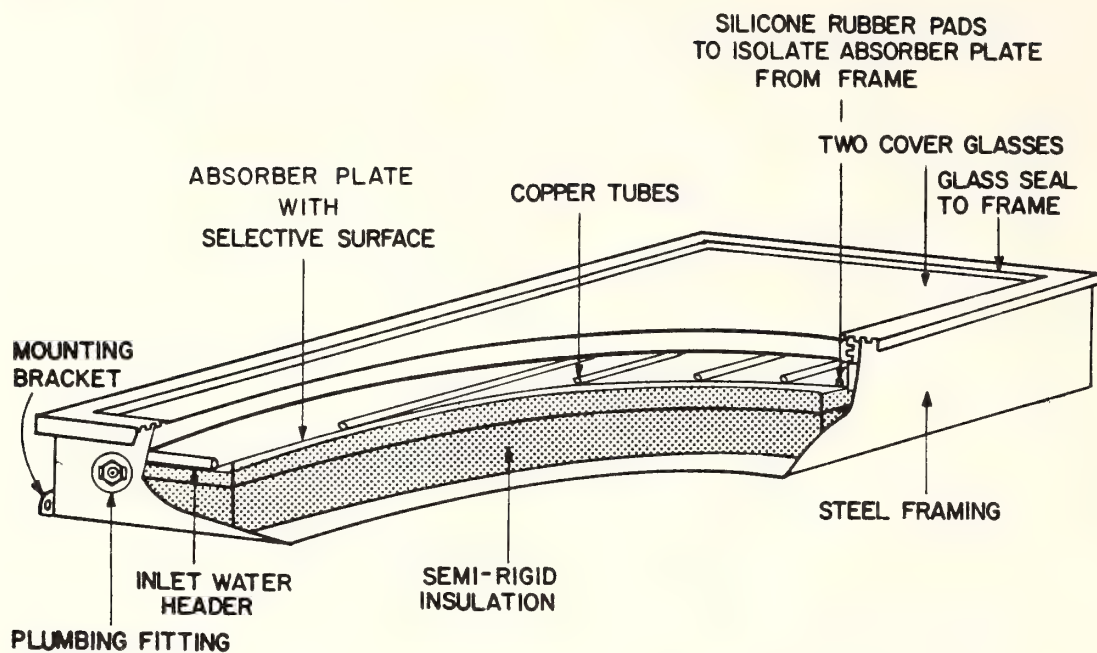


Figure 2-5. Typical Liquid-Heating Collector with Tubular Liquid Passages

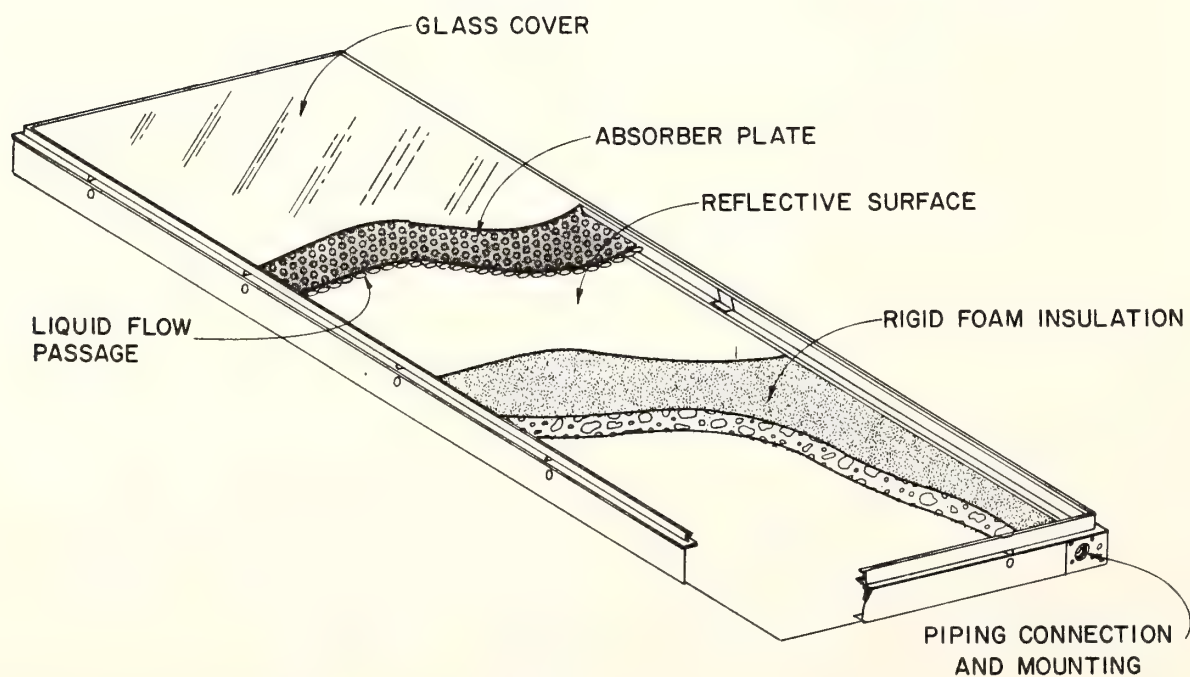


Figure 2-6. Liquid-Heating Collector with Flow Between Metal Plates

Each collector consists of an absorber plate (commonly a blackened metal surface) which absorbs the incident solar radiation and increases in temperature. Heat in the absorber plate then flows to a liquid which circulates through passages in the plate and delivers the heat to other parts of the system. As it collects energy, the absorber plate loses some heat to the surroundings, so the collector is designed to reduce these losses to the minimum practical level.

Heat may be lost upward and downward from the absorber plate by radiation, conduction, and convection. Since the cost of the solar heat supply depends directly on the total collector area required, reduction of heat losses will permit use of less collector area and reduced cost. Insulation beneath the absorber and transparent covers above it reduce all three types of heat loss. Glass is opaque to thermal radiation emitted from the absorber plate, and a glass cover can also reduce convection losses due to air movement across the absorber. An air space between the absorber plate and cover acts to reduce convection losses between these two surfaces.

A special type of flat-plate collector comprises a glass tube surrounding a flat or cylindrical absorbing surface. As shown in Figures 2-7 and 2-8, a high vacuum inside the tube minimizes heat losses from these collectors. There is no concentration of radiation in this type, but efficiencies and delivery temperatures may be considerably higher than usually obtained in conventional flat-plate collectors.

A concentrating or focusing collector gathers solar radiation falling on a large area and focuses the energy onto a smaller absorber area. Concentrating collectors can deliver heat at higher temperatures

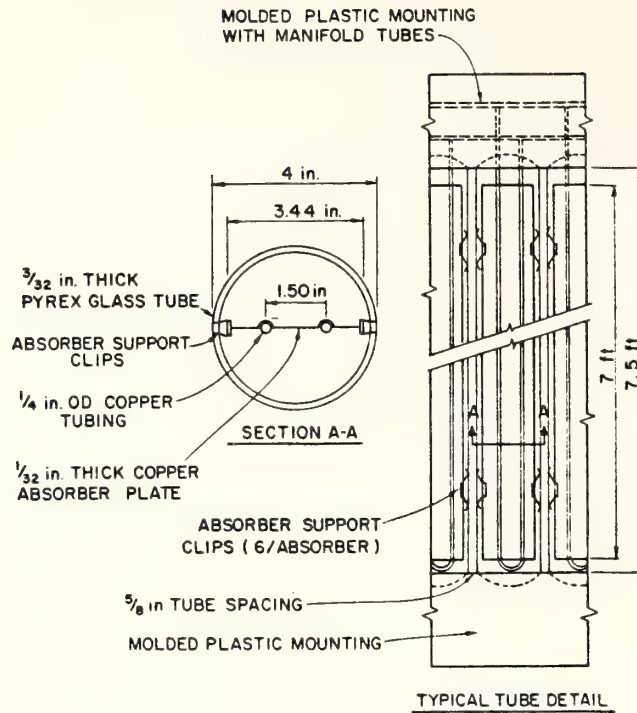


Figure 2-7. Corning Glass Company Evacuated Tube Collector

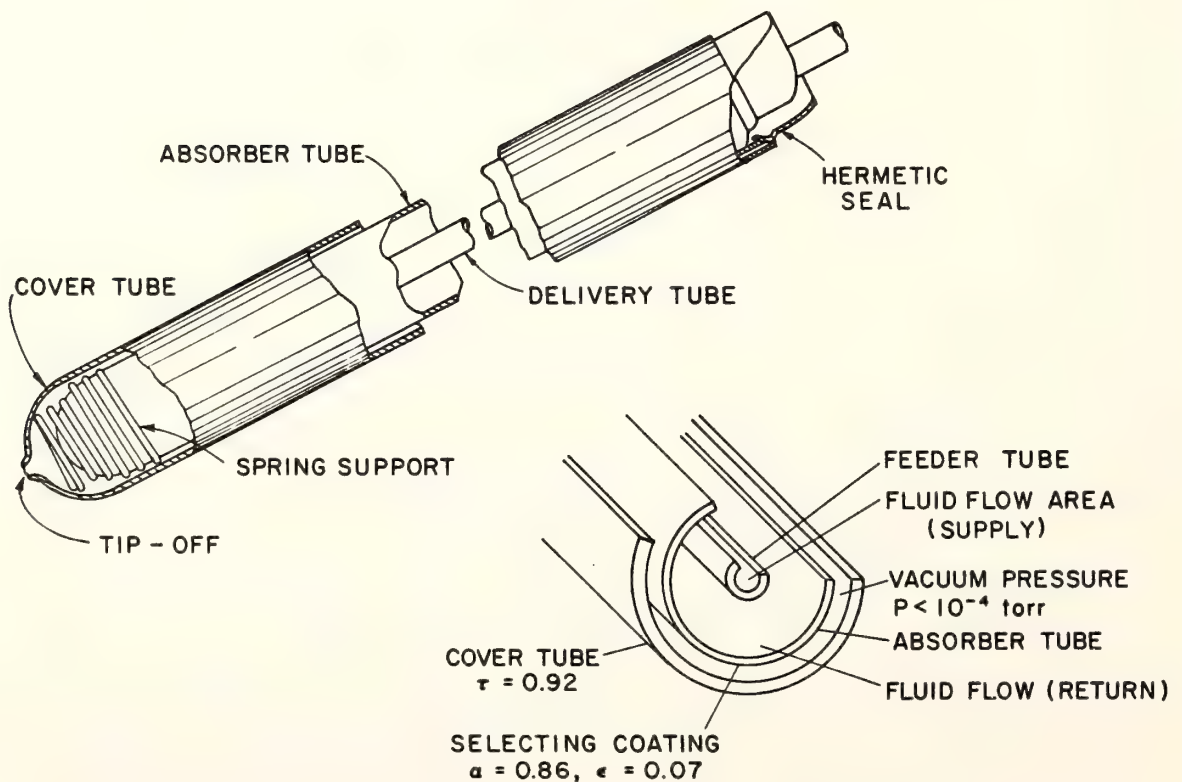


Figure 2-8. Schematic of the Owens-Illinois Evacuated Tube Solar Collector

than flat-plate types and may therefore be used for steam generation and other applications for which such temperatures are necessary. But if this type is used for heat supply at the moderate temperatures needed for space heating and hot water, collector area requirements are comparable to those of the flat-plate type but equipment costs are higher.

Thermal Storage Units

The absence of solar radiation at night and the need for continuous space heating in cold weather, results in a requirement for storing excess solar energy collected, but not needed, during sunny daylight hours. The stored energy can then be used for meeting nighttime heating demands. This energy storage requirement is most economically met by transferring the excess heat to a mass of solid or liquid in an insulated container. Numerous materials could be used, but because of simplicity and economy, most commercial liquid solar systems utilize hot water storage.

It is technically possible to store heat in scrap metal, melted chemicals, waxes, rock, ceramic bricks, and other materials. Water has a higher heat storage capacity than any other material, pound-for-pound, unless melting of a solid is involved. It is also the cheapest heat storage material, but the cost of a container must be considered. Heat can readily be stored in water by increasing its temperature during the day, then using the hot water for heat supply at night.

Several chemical salts and waxes can store heat by melting rather than by increase in temperature. When the molten material again solidifies, the stored heat is released for use. Because the heat stored by

melting a substance is considerably greater than the heat involved in changing the temperature of an equal mass of water fifty degrees F or so, a phase-change heat storage unit can be much smaller than the other types. However, because of economic disadvantages and technical difficulties, phase-change storage materials are not ready for practical use in solar heating and cooling systems.

Extensive studies and experiments have shown that for space heating and domestic hot water supply, practical and economical heat storage units should have a capacity sufficient for storing one day's solar collection for use the following night. Larger storage capacities provide only small additional gains, so the storage of sufficient heat for use during one or more cloudy days is uneconomical.

AIR-HEATING SOLAR SYSTEMS

A BASIC SYSTEM

The main components and their primary functions in an air system are the same as in a liquid collection and storage system. This can be seen by comparing Figure 2-9 with Figure 2-1. An important characteristic of the air system in comparison with the liquid type is the direct supply of heat from the collector to the living space. Whereas liquid collection and storage systems may involve heat distribution either as hot air or as hot water, air is always used for distribution when it is also the collection medium. Auxiliary heat is nearly always supplied as a supplement or boost in air systems (Figure 2-9), but in liquid systems having hot water distribution (Figure 2-1), the auxiliary is in parallel rather than in series with the solar supply.

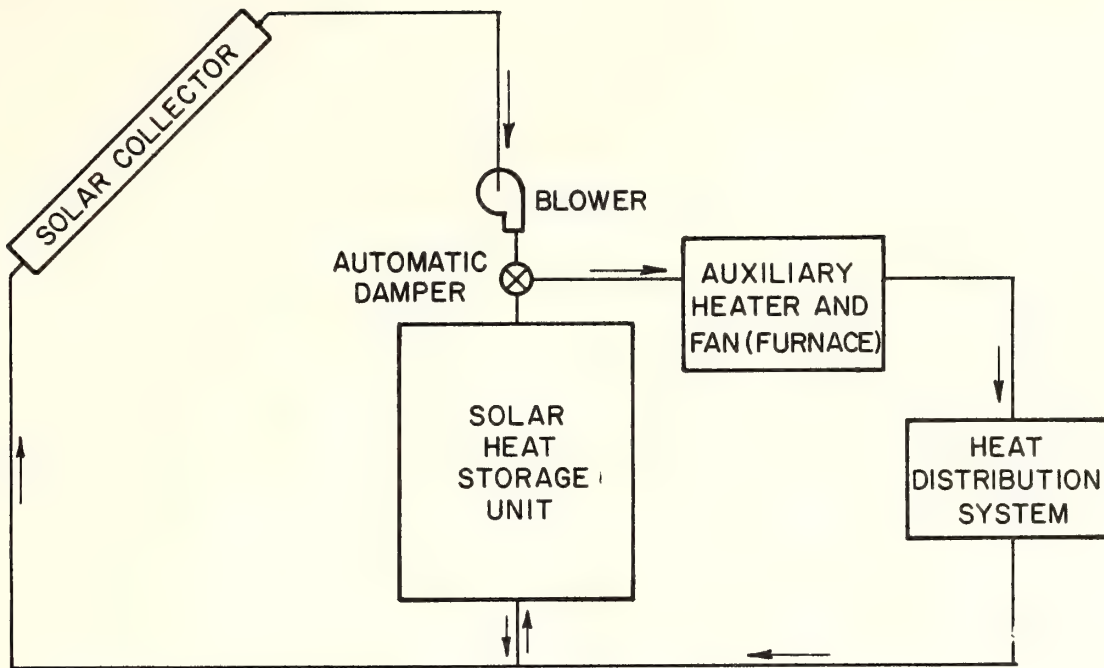


Figure 2-9. Schematic Diagram of an Air-Heating Solar System (Simplified)

The most economical and effective type of heat storage for an air system is a rock-filled bin through which air is circulated and in which heat is transferred to and from egg-sized pebbles or crushed rock. This key component serves not only as a heat storage unit but also as a heat exchanger during the storing cycle and as a heat exchanger during the transfer of stored heat to air being supplied to the rooms. The use of a pebble bed, as shown in a subsequent module, results in beneficial temperature stratification in the storage unit and a substantially higher solar collector efficiency than if transfer to hot water storage were attempted.

SYSTEM OPERATION

A common type of air-heating solar system is shown in Figure 2-10. The components include (1) air-heating solar collector, (2) pebble-bed

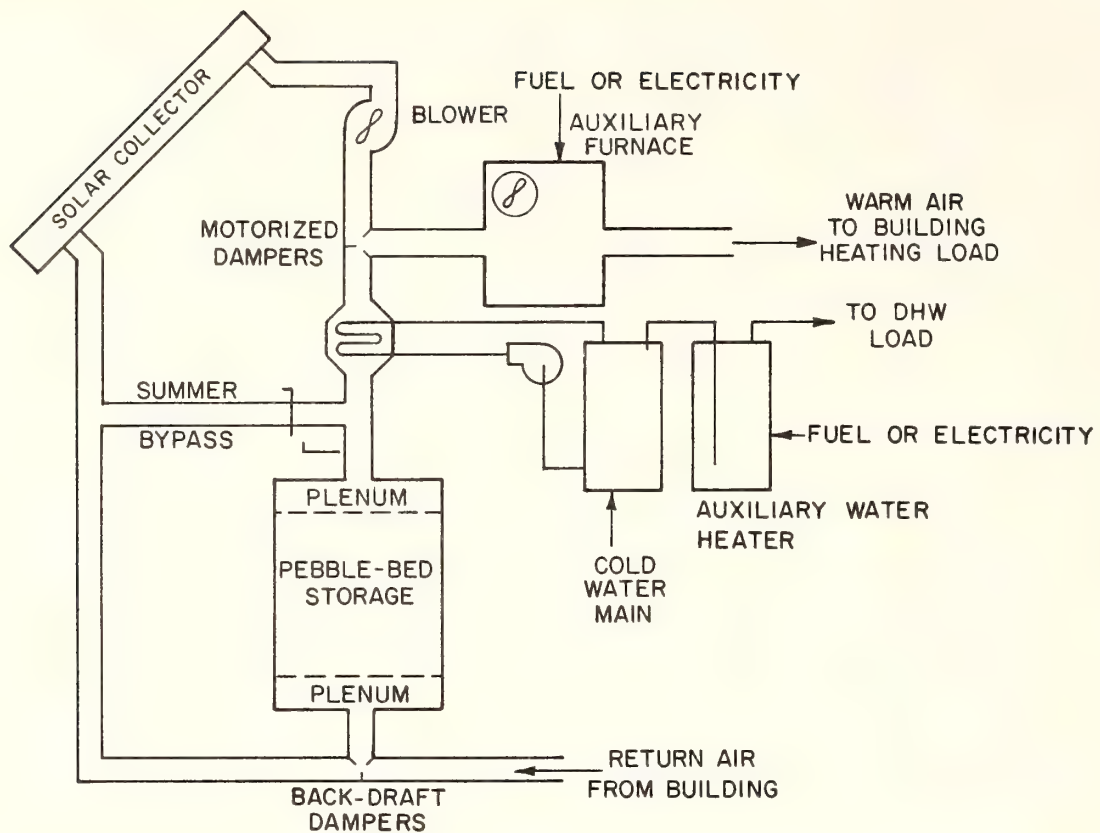


Figure 2-10. Air-Heating Solar System

heat storage unit to and from which heat is transferred by circulating air through the bed, (3) control unit which includes the sensors and control logic necessary to collect and store solar heat when available and to automatically maintain comfort conditions at all times, (4) air handler comprising automatic dampers, filters, and blower, (5) solar hot water heater consisting of an air-to-water heat exchanger and a preheat storage tank connected to an auxiliary hot water heater, and (6) warm-air furnace to provide auxiliary heat when storage or collector temperatures are insufficient to meet demands.

As air flows from one end of the collector to the other, its temperature normally rises from about 70 degrees to about 130 to 150 degrees (F) during the mid-part of the day. Whenever heating is needed

during sunny periods, warm air is supplied to the rooms directly from the collector via the auxiliary furnace. Cool air from the building is returned to the collector for reheating.

When heat is not needed in the building, solar-heated air is routed through the storage unit, thereby heating the pebbles; cool air, usually at 70°F, returns from the pebble bed to the collector for reheating. Temperature stratification in the pebble bed assures maximum heat recovery from the collector.

When the sun is not shining, heat is delivered to the rooms by circulating air from the building through the pebble bed. Because of temperature stratification in the storage unit, this mode supplies heat to the rooms at the highest available temperature. The system automatically provides auxiliary heating from fuel or electricity when the solar supply is insufficient to meet requirements.

SERVICE WATER HEATING

Service hot water can be heated by use of an air-to-water heat exchanger in the hot air duct from the collector. When solar-heated air is being delivered to storage, heat is also being supplied to the hot water system by operation of the water pump. The temperature of the air passing through the water heating coil is thus usually reduced by a degree or two. Other coil locations have been used, but this position has proved to be the most satisfactory. So that solar-heated water can be available in the summer when no space heating is needed, a by-pass duct is opened and air returns to the collector without passage through storage or the rooms.

SYSTEM COMPONENTS

Solar Collectors for Air Heating

Solar air collectors resemble the liquid heating types and differ primarily in the design of the fluid passages in contact with the absorber plate. There may also be a difference in absorber plate materials, steel or aluminum normally being used in air collectors, whereas corrosion considerations usually dictate use of copper or stainless steel in liquid types. Figures 2-11 and 2-12 show two commercial types of air collectors, one featuring internal manifolds for air distribution to and from an array of numerous panels, and the other employing a perforated absorber surface. Glazing, bottom insulation, and metal box are similar to those used in the liquid types.

Numerous variations in the design of solar air collectors have been used. Corrugated, finned, perforated and other modified absorber plate forms have been tested. The most widely used type of air collector contains a smooth black absorber plate beneath which air is circulated in a space about one-half inch high.

Heat Storage in Air Systems

Because of low cost, high heat transfer effectiveness, and widespread availability, gravel or crushed rock is nearly always used for heat storage in air-heating solar systems. An insulated rectangular bin is usually constructed in a basement or a utility area in the building. The bin walls may be of concrete, masonry, or wood, and the top is usually of wood. Clean aggregate normally used in concrete, screened to a 3/4-inch to 1½-inch size range, is supported on wire mesh above spaced concrete blocks or steel frames on the bin bottom. Depths of 5 to 7

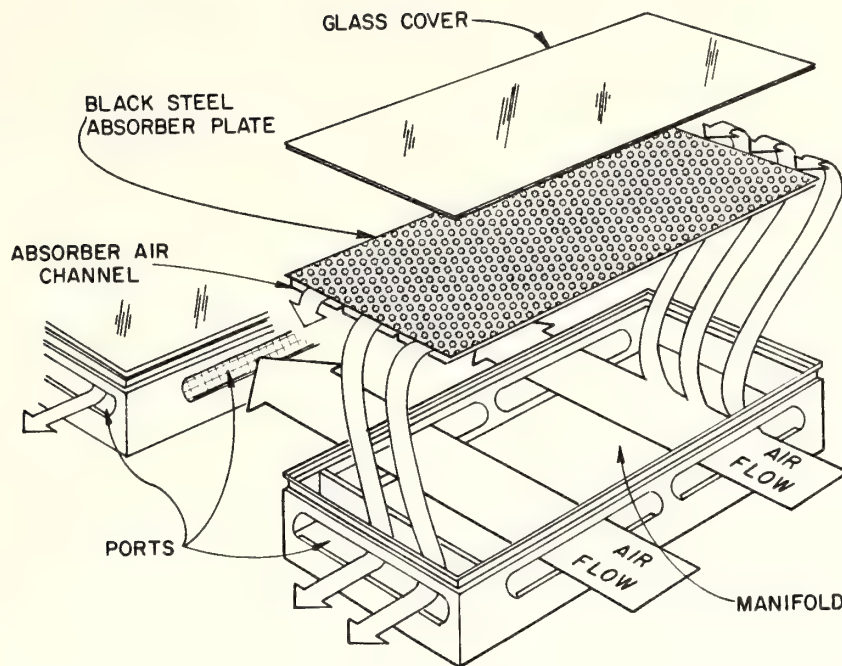


Figure 2-11. Air Collector with Ducts Beneath Absorber and Internal Manifolds (Exploded View)

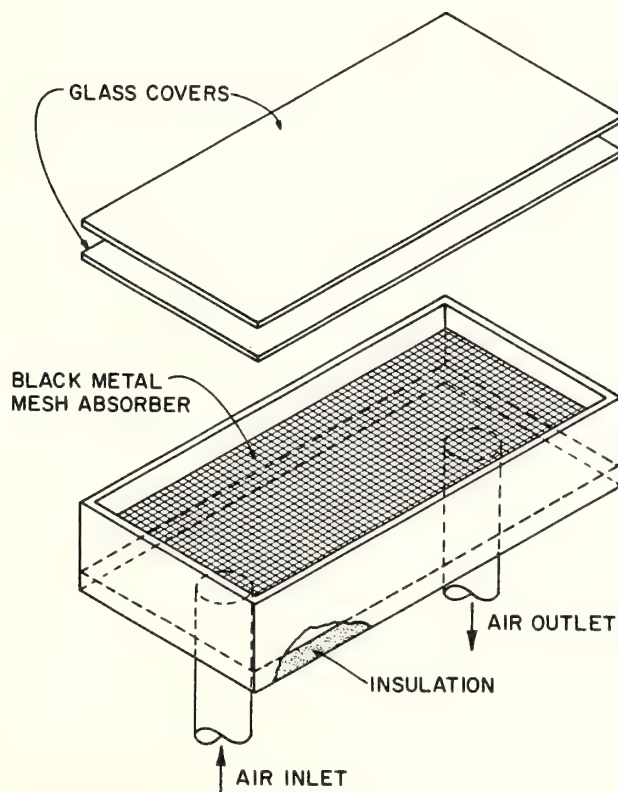


Figure 2-12. Air Collector with Perforated Absorber Plate (Exploded View)

feet are typical, and air supply and withdrawal openings are provided in the top and bottom plenum spaces. Figure 2-13 shows a cut-away view of a pebble bed in a wood bin.

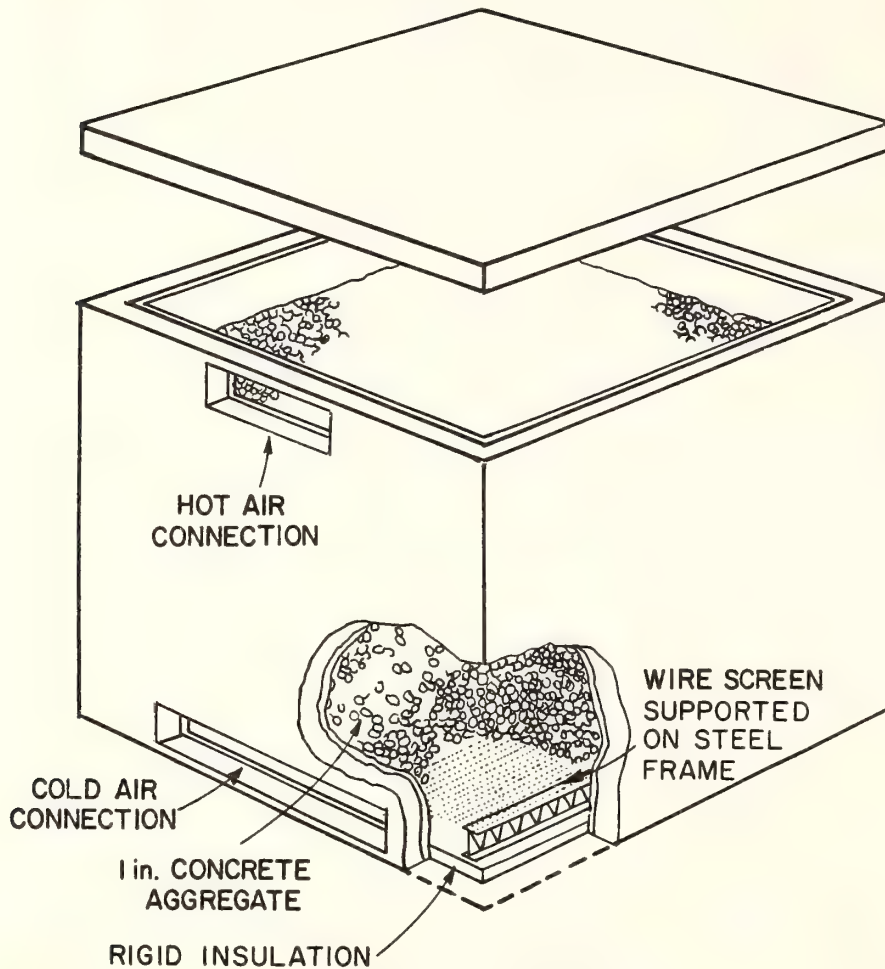


Figure 2-13. Pebble-Bed Heat Storage Unit

Heated air supplied to the top of a pebble bed from a solar collector passes down through the bed and leaves the bottom essentially at the rock temperature in the lowest portion of the bed. Air returning to the collector has thus been fully cooled and its useful heat content has been transferred to pebbles usually in the upper part of the bed.

AUXILIARY HEATERS

HYDRONIC SYSTEMS

During cloudy periods and on mid-winter nights, the solar system is usually unable to meet all of the heat needs of the building. In a liquid system, a conventional hot water boiler (Figure 2-14) may be provided to supply part or all of the heating requirements during these periods. If the temperature in the solar storage tank is too low to maintain the preset building temperature, the auxiliary boiler automatically supplies hot water to the distribution system. When a hot water boiler is used for auxiliary heat supply, it is used in parallel, not in series, with the solar supply. It is therefore an alternate heat source rather than a "booster", so that auxiliary will not partially feed back to solar storage.

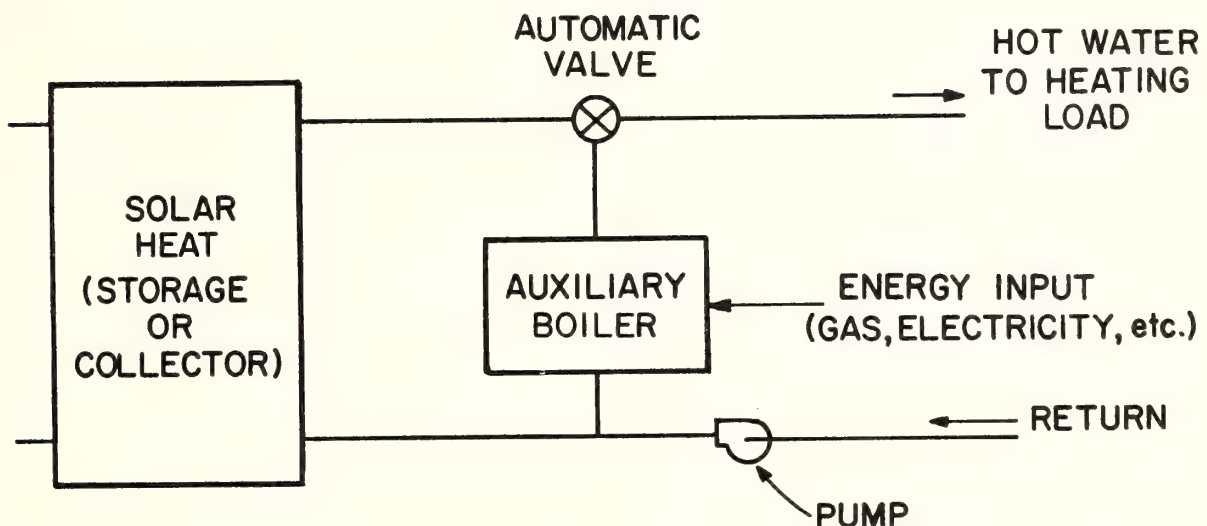


Figure 2-14. Typical Use of Auxiliary in Liquid (Hydronic) Distribution System

AIR FURNACES

If the building is provided with a warm-air heating system, and if solar heat is supplied from either liquid or air collectors, an air furnace is usually used for auxiliary heat. Figure 2-10 shows auxiliary use in all-air systems, and Figure 2-15 shows a liquid-to-air heat exchanger and warm-air auxiliary furnace in a solar liquid system. Electricity or any type of fuel may be supplied to the furnace. Auxiliary heat may also be provided by a heat pump.

Although the auxiliary heater is usually called upon to supply only part of the total space heating demand, there will be occasions when no solar heat is available from the collector or storage. Full design capacity must therefore be provided so that comfort can be maintained without solar supply during the coldest weather.

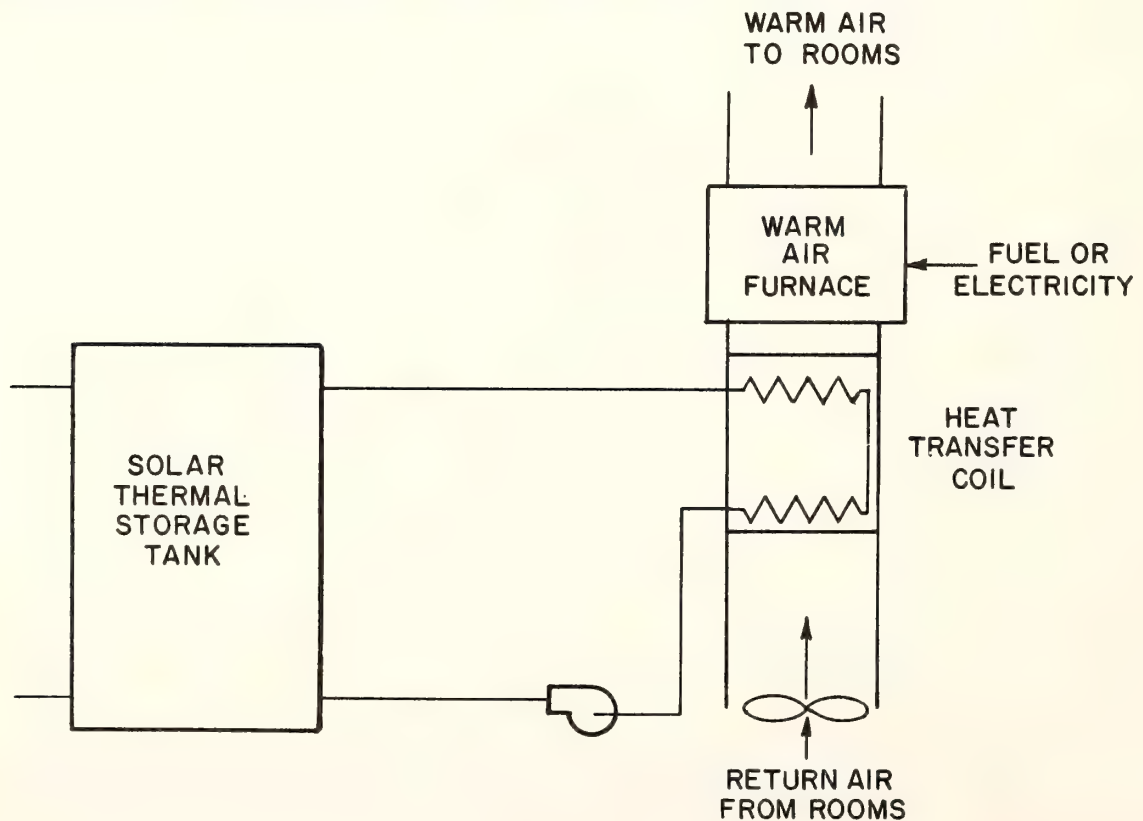


Figure 2-15. Solar Heat from Liquid System to Warm Air Distribution System

HEAT DISTRIBUTION

In "hydronic" systems, hot water can be piped from storage or auxiliary boiler to coils imbedded in floors or ceilings (radiant heating) or to "radiators", fan-coil units, or baseboard strip heaters in individual rooms. The operating temperature of most baseboard hot water heaters is about 180°F, which is too high for use with typical solar systems. But if additional heating surface is provided, as for example, with double rows of baseboard tubes, lower temperatures may be usefully employed, and solar-heated water can be supplied at suitable temperatures.

Forced warm air distribution is commonly used with all types of solar heating systems. In liquid collection and storage systems, hot water from the solar storage tank is pumped to a heating coil (finned-tube exchanger) as illustrated in Figure 2-15, and circulating air is heated as it passes through the coil. Figure 2-10 shows that in all-air systems, warm air from collector or storage is delivered through conventional distribution ducts via the auxiliary furnace in which additional heat is supplied if needed.

AUTOMATIC CONTROLS

To control the temperature in a conventionally heated home, the homeowner needs only to set a thermostat. The same is true in a house with a well-designed solar heating system. However, controls for solar heating and cooling are more complex than in a conventional system, because they must operate collector and storage pumps and blowers,

automatic valves and dampers, as well as the conventional equipment. To illustrate the principles, a schematic diagram of a control assembly for a liquid-heating solar system for space and domestic water is shown in Figure 2-16.

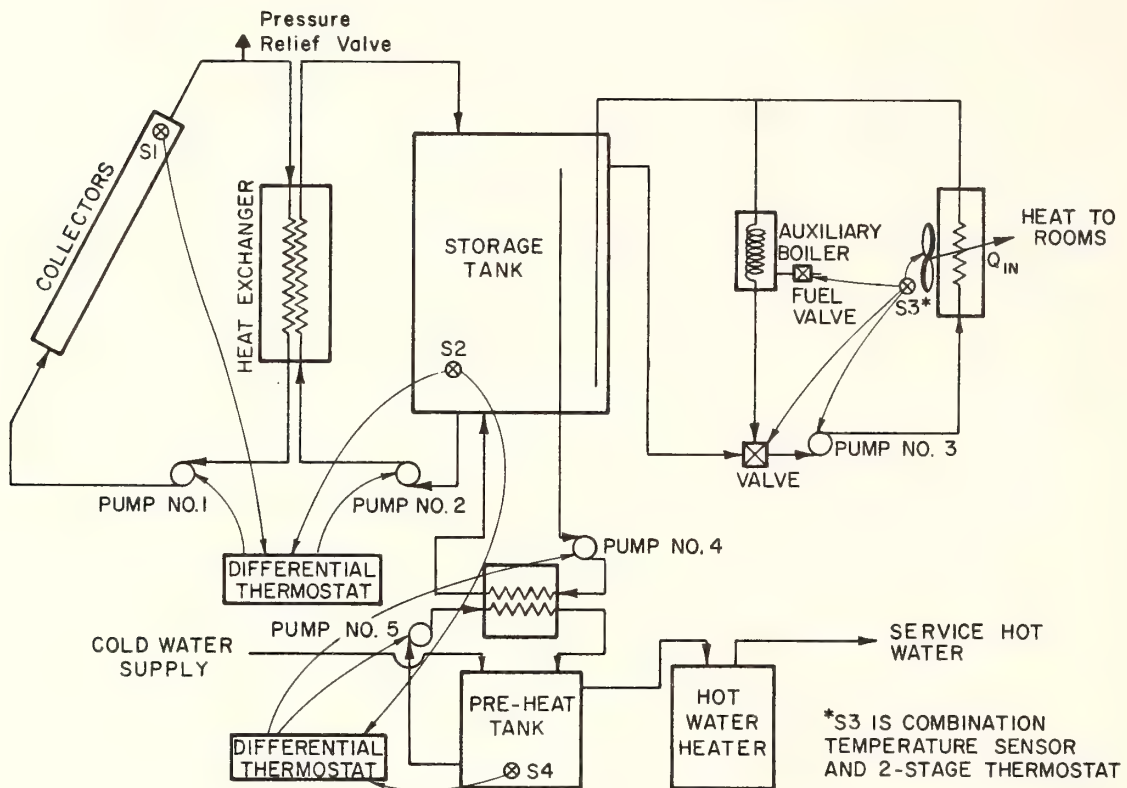


Figure 2-16. Typical Controls for a Liquid-Heating Solar System

In nearly all solar heating systems, a differential thermostat senses the difference in temperature between collector outlet and storage. When this difference is more than a preset number of degrees, the circulating pumps are operated and solar heat is stored. A pressure relief valve protects the system from excessive pressure which might otherwise develop if there is circulation failure during a sunny period.

Supply of heat to a building is usually controlled by a two-stage thermostat, the normal setting actuating pumps and fans which circulate solar heated air or water to the rooms. If the room temperature continues to drop, a second (lower temperature) contact point in the thermostat actuates the supply of fuel or electricity to the auxiliary heater.

Preheating of service water is controlled by a differential thermostat which senses the difference in temperature between the solar storage tank and the preheat tank. At a difference of more than a few degrees, circulation pumps are operated and heat is transferred from the main storage to the preheat tank. A "high set" thermostat (limit control) may be used to prevent too high a temperature in the preheat tank by interrupting power to the circulation pumps.

Numerous controllers for solar space heating systems are commercially available, and many varieties of control circuits and methods are being used. Design of control systems requires directions from the manufacturers of the control components and experience in their proper integration and adjustment.

SOLAR COOLING

There are several possible methods for cooling with solar energy. These include absorption refrigeration (lithium bromide-water and ammonia-water systems), Rankine-cycle vapor-compression, and desiccant-evaporative cooler combinations. Only the lithium-bromide absorption type is commercially available, and its operation with solar heat has been almost entirely in experimental installations. Although not solar

operated, heat pumps are sometimes used in solar heating systems for auxiliary heat supply and for conventional cooling with electric power.

An absorption cooling system in which solar-heated water is the energy source is illustrated in Figure 2-17. Solar cooling by this and other methods is complex and expensive, so it is currently in an experimental status. Cooling machines that can be operated with hot water at 160°F to 200°F are commercially available in sizes from 3 tons to over 25 tons of cooling capacity. Chilled water, at 40°F to 45°F, is produced and supplied to one or more fan-coil units in the air circuit. Air is cooled, dehumidified, and distributed to the rooms. Heat is rejected from the system in cooling water which is recirculated through a cooling tower.

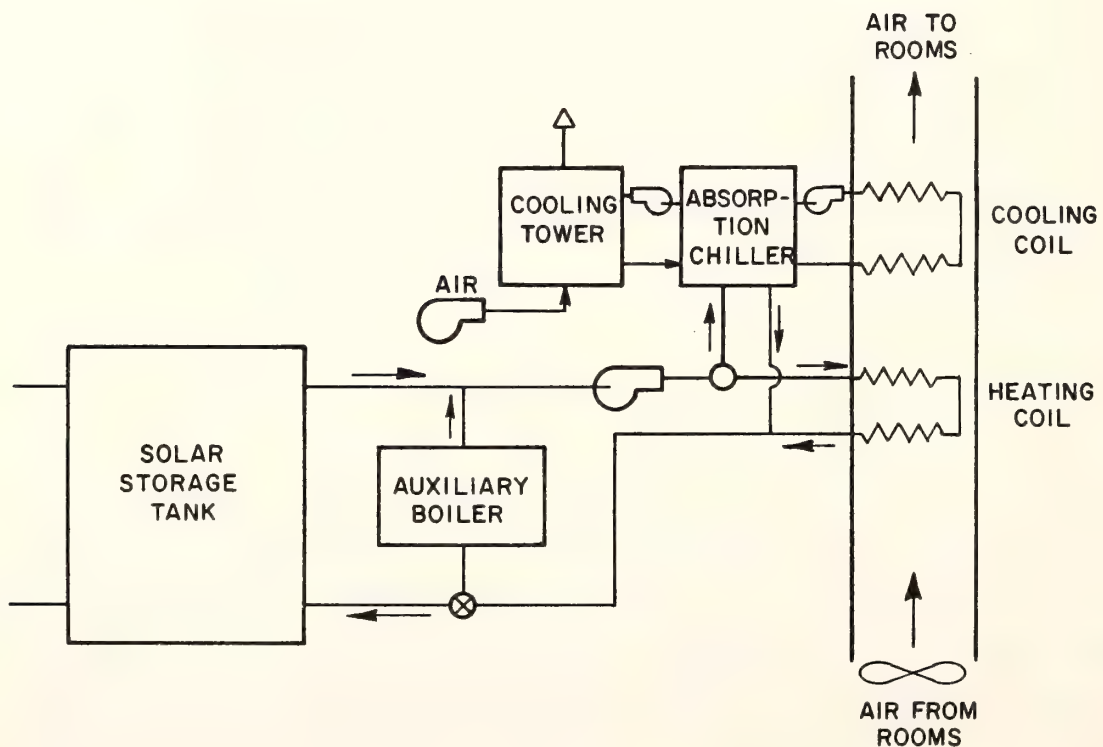


Figure 2-17. Solar Heating and Cooling System

Other experimental methods of solar cooling include conventional vapor compression driven by power generated in an on-site engine supplied with steam or organic vapor from a solar collector/boiler. Desiccant (air-drying) systems in which room air is dehumidified and the solid or liquid drying agent is regenerated by solar heat are also being investigated. None of these systems is at present technically or economically practical.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 3

SOLAR RADIATION

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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GLOSSARY OF TERMS

Azimuth	Deviation in angle from due south.
Beam Radiation	Radiation reaching the earth's surface from the solar disc.
Btu	British Thermal Unit, the quantity of heat required to raise the temperature of one pound of water one degree Fahrenheit.
Calorie	The quantity of heat required to raise the temperature of one gram of water one degree Centigrade.
Declination	Position of the sun relative to the equatorial plane at solar noon.
Diffuse Radiation	Radiation reaching the earth's surface from 180 degree hemisphere, excluding the beam radiation.
Direct Radiation	Radiation reaching the earth's surface from the solar disc.
Extraterrestrial Radiation	Solar radiation outside the earth's atmosphere.
Horizontal Surface	Surface that is tangent to the earth's surface at any point, and any surface that is parallel to the tangent surface.
Langley	Measure of solar radiation intensity.
Solar Constant	Solar Radiation intensity on a surface normal to the sun's rays at the mean earth-sun distance, 428 Btu/(hr·ft ²)
Solar Radiation	Portion of total radiation which is useful for solar heating and cooling systems.
Solar Radiation Flux	Time rate of delivery of solar radiation.
Tilted Surface	Tilted with respect to the horizontal.

LIST OF SYMBOLS

\bar{D}	Average daily diffuse radiation for a month
\bar{I}_H	Monthly average daily total radiation on a horizontal surface, Btu/(day·ft ²)
I_{sc}	Solar constant, 428 Btu/(hr·ft ²)
I_o	Extraterrestrial radiation on a horizontal surface at the outer limits of earth's atmosphere, Btu/(hr·ft ²)
\bar{I}_T	Monthly averaged daily radiation on a tilted surface, Btu/(day·ft ²)
\bar{K}_t	Fraction of solar energy which penetrates through the earth's atmosphere on daily average
n	Number of days from January 1
\bar{R}	Fraction of average daily radiation on tilted surface compared to a horizontal surface
R_D	Ratio of the average daily beam radiation on a tilted surface to that on a horizontal surface
s	Collector tilt angle from horizontal, degrees
T	Temperature, °F
t	Time variable
ω_s	Sunset hour angle, degrees from solar noon
δ	Solar declination, position of the sun relative to the equatorial plane at solar noon, degrees
ϕ	Latitude angle, degrees (north is plus)
ρ	Reflectivity of material or ground surface

OBJECTIVES

The objective in this module is to present methods for calculation of solar radiation incident on a tilted collector surface by use of solar radiation data on a horizontal surface. With the information in this module the trainee should be able to:

1. Recognize the factors which affect the variability of solar radiation at the earth's surface.
2. Estimate mean daily solar radiation for each month of the year.
3. Convert solar radiation data from one set of units to another.
4. Determine the solar radiation intensity on a collector surface at any tilt and orientation.

INTRODUCTION

Energy is radiated from the sun as a result of thermonuclear reactions which take place within its core. Violent action that accompanies the thermonuclear reactions leads to particulate emission from the sun and the high surface temperatures result in electromagnetic radiation (or more simply, radiant energy) emission outward through space. The particulate emission consists of electrons and protons and is commonly called "solar wind". The electromagnetic radiation is commonly termed "solar radiation" and is the form of energy with which we are concerned in this module.

The electromagnetic radiation consists of a wide spectrum of wavelengths and energetic intensities as shown in Figure 3-1. Almost half of the solar energy received on earth is in the band of visible light,

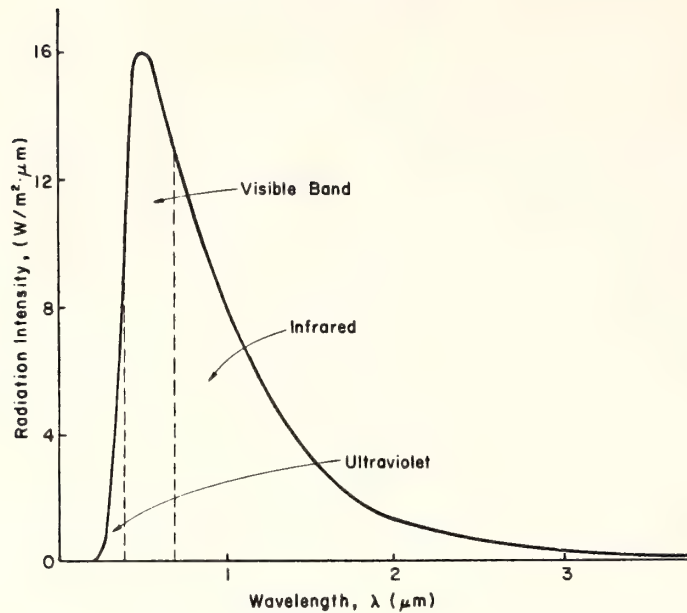


Figure 3-1. Spectrum of Solar Irradiance at the Earth's Surface

and nearly all of the other half consists of the near infrared wavelengths. A small portion is attributable to ultraviolet and other short wave length energy radiation, most of which is filtered out by the earth's atmosphere. The intensity of solar energy varies with latitude, altitude, time of day and time of year. Figure 3-1 shows a "smoothed" spectral distribution (narrow absorption bands due to carbon dioxide, water vapor, and ozone in the atmosphere have been averaged into nearby wavelengths) of solar energy received at sea level, under "standard" atmospheric conditions known as air mass 1.0.

SOLAR CONSTANT

The solar radiation flux (time rate of delivery of solar energy) varies inversely with distance from the sun, and since the earth is in an elliptical orbit about the sun, the energy flux reaching the outer limits of the earth's atmosphere varies from about 410 to 440 Btu/(ft²·hr). At the mean earth-sun distance, the energy flux is called the

solar constant, I_{sc} , and its magnitude is 428 Btu/(ft²·hr), equal to 1.35 KW/m².

EXTRATERRESTRIAL RADIATION

Solar radiation at the outer limits of the earth's atmosphere is called extraterrestrial radiation, I_o . The extraterrestrial radiation on a plane which is parallel to, and directly above, a horizontal plane on the earth's surface will vary with respect to latitude and time of year. The monthly average daily extraterrestrial radiation, \bar{I}_o , on a horizontal surface is calculated for each month at various latitudes and presented in Table A3-1 in the appendix to this module.

VARIATIONS IN TERRESTRIAL RADIATION

As the solar rays penetrate the earth's atmosphere, radiation is scattered, absorbed, and reflected by the atmospheric constituents such as carbon dioxide, oxygen, ozone, water vapor and particulate matter in the atmosphere as shown in a schematic representation in Figure 3-2. Some of the energy is reflected back into outer space at the outer fringes of the atmosphere and still more is reflected from the tops of clouds. As much as 30 percent of the extraterrestrial radiation may not reach the earth's surface on a clear day. Solar radiation which penetrates the atmosphere without being reflected or scattered is called direct, or beam radiation. Diffuse radiation reaches the earth's surface by reflection from particulate matter and clouds in the atmosphere. At a particular point on the ground, therefore, diffuse radiation arrives from clouds and the entire sky.

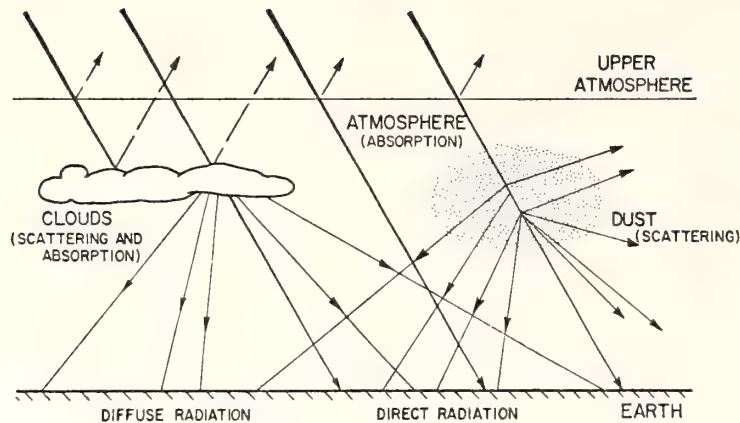


Figure 3-2. Atmospheric Effects on Solar Radiation Reaching Earth

MONTHLY VARIATIONS

The mean daily radiation on a horizontal surface, \bar{I}_H , varies from month-to-month and with geographic location, due to seasonal changes in weather and changing angular relationship between the sun and earth's surface. In the winter, the sun is lower in the sky than in the summer, and the resulting smaller angle between the sun and a horizontal surface reduces the amount of radiation intercepted by the surface, as shown in Figure 3-3(A). In the summer, the sun is higher above the horizon (at noon) than in winter and a larger amount of energy is intercepted by a horizontal surface as shown in Figure 3-3(B).

An example of the variation in the monthly average daily radiation on a horizontal surface is shown in Figure 3-4 for Boulder, Colorado. It is noted that the average daily radiation in June and July is about twice that in January.

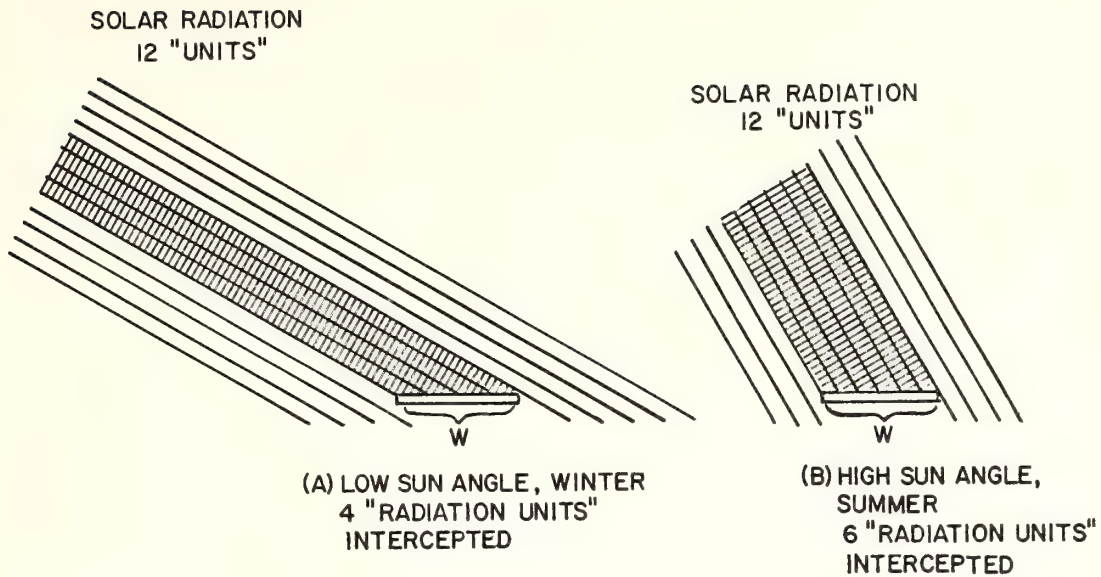


Figure 3-3. Energy Intercepted by a Unit-Width Horizontal Surface

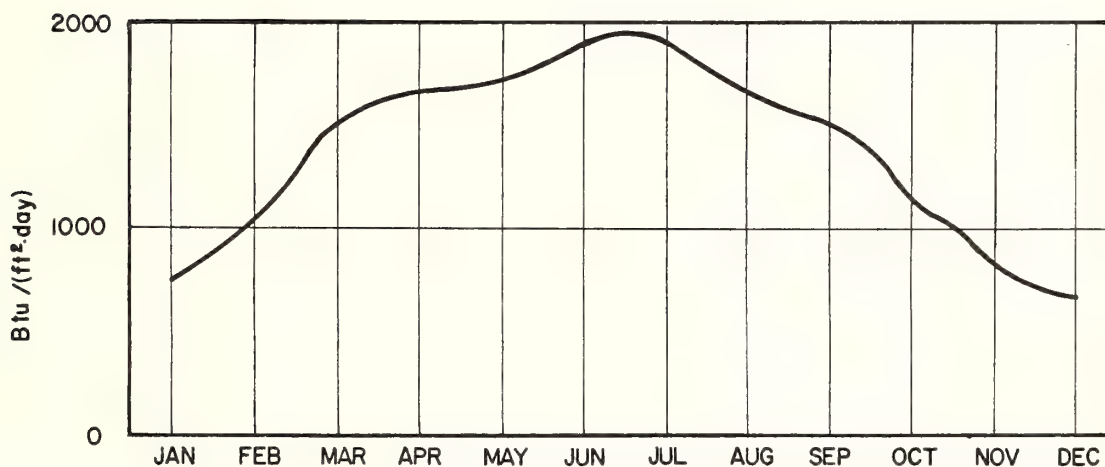


Figure 3-4. Monthly Variation of Average Daily Radiation on a Horizontal Surface, Boulder, Colorado (From the Climatic Atlas of the United States)

DAILY VARIATIONS

There are wide variations in total daily radiation on a horizontal surface largely as a result of clouds. On overcast days when the total radiation is largely diffuse, there may be only two or three hundred

Btu/ft² for the entire day with the intensity so low that a solar system could not collect useful heat. On sunny days, two to three thousand Btu/ft² may be received.

HOURLY VARIATIONS

Hourly variations in total solar radiation are a result of the earth's rotation about its own axis. Early morning sun is at a very low angle and the solar rays must penetrate a "thick" atmospheric layer. Maximum radiation occurs at noon, when the sun is at the highest angle above the horizon and the radiation encounters a minimum thickness of the atmosphere.

Hourly variations of solar radiation on a horizontal surface, measured in Fort Collins, Colorado, on clear days are shown in Figure 3-5. Clouds would, of course, create random variations in solar intensity and the total radiation would be less than on clear days.

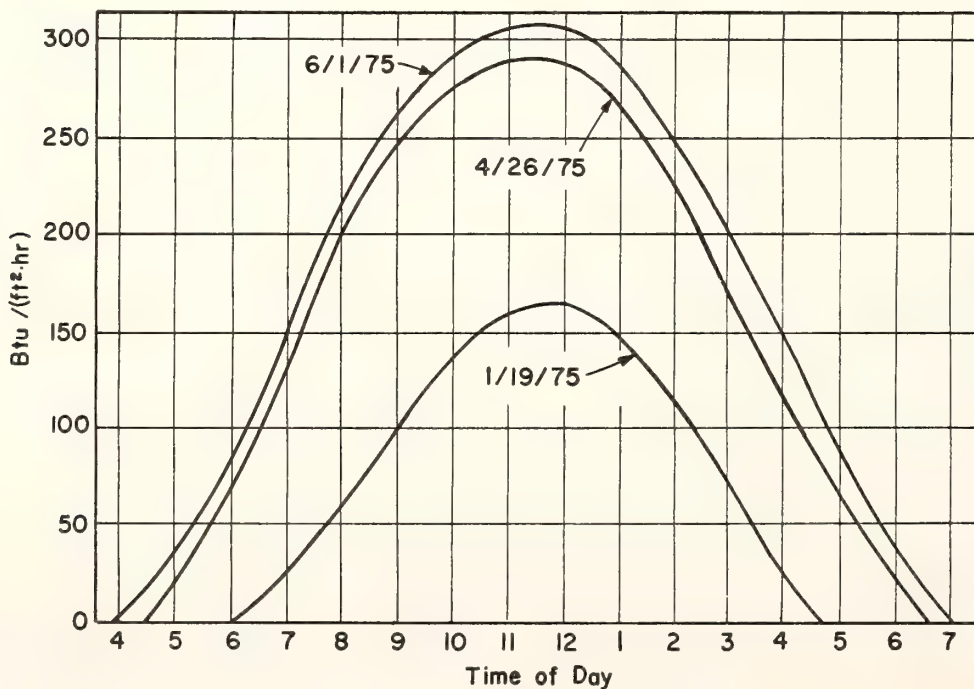


Figure 3-5. Hourly Record of Total Solar Radiation on a Horizontal Surface on Clear Days in Fort Collins, Colorado (Data from Solar House I)

REGIONAL VARIATIONS

To emphasize regional variations of solar radiation, the monthly average daily solar radiation at selected cities in the United States is listed in Table 3-1 for four months which represent the four seasons of the year. The variations are due to different latitudes as well as to weather conditions.

Table 3-1

Monthly variations in Energy on a Horizontal Surface
for Selected Cities in the United States
[Btu/(ft²·day)]

City	Latitude	December	March	June	September
Chicago, IL	41°	354	838	1690	1155
Tucson, AZ	32° 07'	1172	1993	2601	2122
Washington, DC	38° 51'	632	1255	2081	1446
Miami, FL	25° 47'	1292	1829	1992	1647
Fairbanks, AK	64° 49'	66	860	1971	700
Los Angeles, CA	34° 03'	912	1641	2259	1892

SOLAR RADIATION DATA FOR DESIGN PURPOSES

DIMENSIONS AND CONVERSION FACTORS FOR ENERGY AND POWER

The intensity of solar radiation is expressed in several different dimensions. In this manual, British Thermal Units (Btu), foot, and hour are used, but other dimensions may be encountered. Thus, the relationships between units are given in Table 3-2 and conversion factors are listed in Table 3-3.

Table 3-2
Energy and Power Units

Abbreviation	Unit
<u>Energy Density</u>	
Btu/ft ²	British Thermal Units per square ft
kJ/m ²	Kilojoules per square meter
Langley (cal/cm ²)	Calories per square centimeter
<u>Power</u>	
Btu/(ft ² ·hr)	British Thermal Units per square ft per hour
kJ/(m ² ·hr)	Kilojoules per square meter per hour
Langley/min	Calories per square centimeter per minute
W/m ²	Watts per square meter

Table 3-3
Conversion Factors for Energy and Power

To Convert into Btu/ft ²		To Convert into Btu/(hr·ft ²)	
<u>Multiply</u>	<u>By</u>	<u>Multiply</u>	<u>By</u>
Langleys	3.69	Langleys/min	221
kJ/m ²	.088	kJ/m ² ·hr	.088
		W/m ²	.316

HORIZONTAL RADIATION DATA

The principal source of solar radiation data for the United States is the National Weather Service Climatic Data Center at Asheville, North Carolina. Presently there are 86 stations throughout the United States and West Indies that are recording total radiation on a horizontal surface. At five stations the direct component is also measured. Of the 86 stations, 67 have their data processed for daily total only, and hourly data are processed at 19 stations. The estimated errors in the recorded data range from ± 5 percent to ± 30 percent, as reported by Jessup (Ref. 1).

Monthly average daily total radiation on horizontal surfaces (from Ref. 2) for 81 locations in the United States and Canada is listed in Table A3-2 of the Appendix. Using these data, maps showing the monthly average daily radiation on a horizontal surface for each month of the year were drawn by the National Weather Service. The maps, printed in the Climatic Atlas of the United States, are reproduced in Figures A3-1 through A3-12 for the months of January through December respectively. The radiation units on the maps are Langleys per day and should be multiplied by the conversion factor, 3.69, to change the units to Btu/(ft²·day).

RADIATION ON TILTED SURFACES

When designing solar systems, it is advantageous to tilt the collectors to be perpendicular to the sun's rays. The increased amounts of solar radiation intercepted by a collector that is tilted, compared to the same collector in a horizontal position, are illustrated in Figure 3-6. When the collector is perpendicular to the incoming radiation, the additional energy intercepted is a maximum as seen in Figure

3-6(B). For any other angle, less solar energy is intercepted as seen in Figure 3-6(C). A collector which follows the sun across the sky so that the rays are always perpendicular to the surface intercepts the maximum amount of energy during the day. However, for many types of collectors, tracking is not practical, and is particularly unsuitable for residential solar heating systems. An alternative arrangement to a tracking collector is a collector array at a fixed tilt to intercept the maximum amount of radiation during a selected period of time, say September through May for a space heating system or during the entire year for a domestic water-heating system.

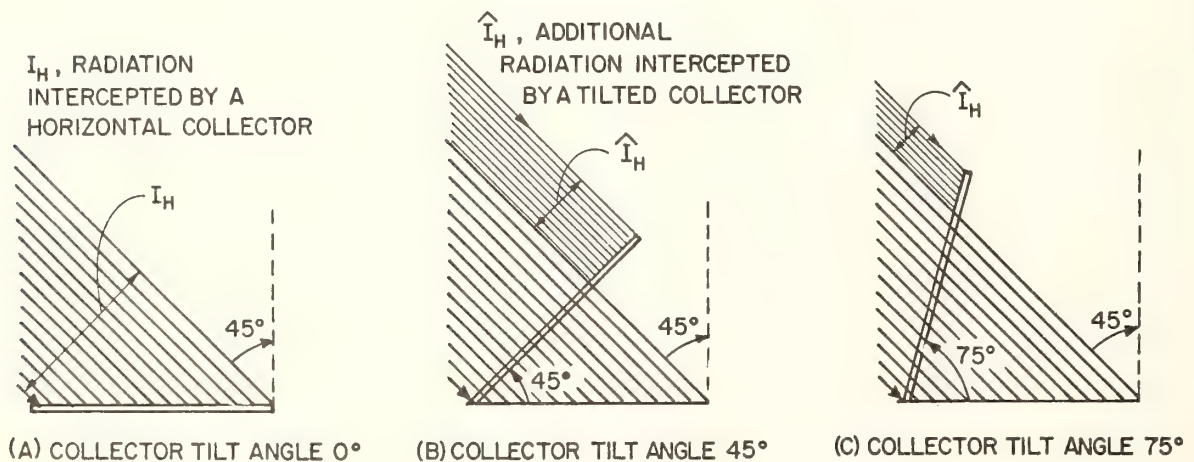


Figure 3-6. Effect of Tilted Surface on Energy Intercepted

To maximize solar energy collection during the heating season, the plane of the collector should be tilted at an angle greater than latitude. The reason is illustrated in Figure 3-7. The declination of the

sun, which is the angle between the plane of the equator and the sun at solar noon, δ , varies from zero degrees on September 21 and March 21, to $-23^\circ 45'$ on December 21 and $+23^\circ 45'$ on June 21, as shown in Figure 3-7(a). Thus a collector tilt, s , which is greater than the latitude angle, ϕ , is more nearly perpendicular to the solar rays from September through March as illustrated in Figure 3-7(b). To maximize summer collection, the collector should be tilted at an angle less than latitude, and if collection is desired throughout the year, a tilt angle nearly equal to the latitude is appropriate. The exact tilt angle for a given location will depend on climatic conditions, but a general rule is to tilt the collectors at latitude plus 15 degrees for a space heating system and at the latitude angle for a domestic water heating system.

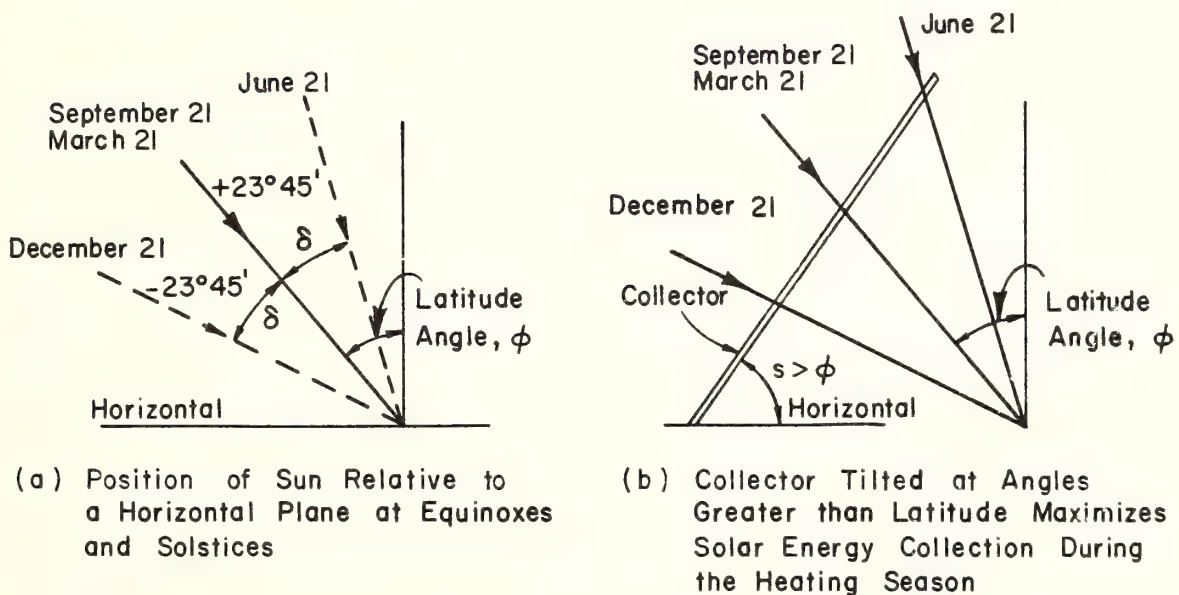


Figure 3-7. Variation of the Sun's Position Relative to Horizontal and Tilted Surfaces

The total solar radiation intercepted by a tilted collector is not only the direct beam radiation as implied in Figure 3-7. There may be a significant amount of diffuse radiation available depending upon climatic conditions and the reflectivity of surfaces in "front" of the collectors. Although a portion of the hemispherical diffuse radiation is "cut off" by a tilted collector, reflected radiation from the foreground may more than offset the reduction of total diffuse sky radiation.

EFFECT OF COLLECTOR ORIENTATION

Because the arc of the sun is symmetrical about solar noon, the preferred orientation of the collector is due south. Any other collector orientation (non-zero azimuth angle) will decrease the total energy incident on the collector surface during the day. However, deviations either east or west by as much as 30 degrees (two hours) will decrease the total daily solar radiation by less than 5 percent as shown in Figure 3-8. An azimuth angle larger than 30 degrees will noticeably affect the solar system. In Figure 3-8, s_0 is the optimum tilt angle.

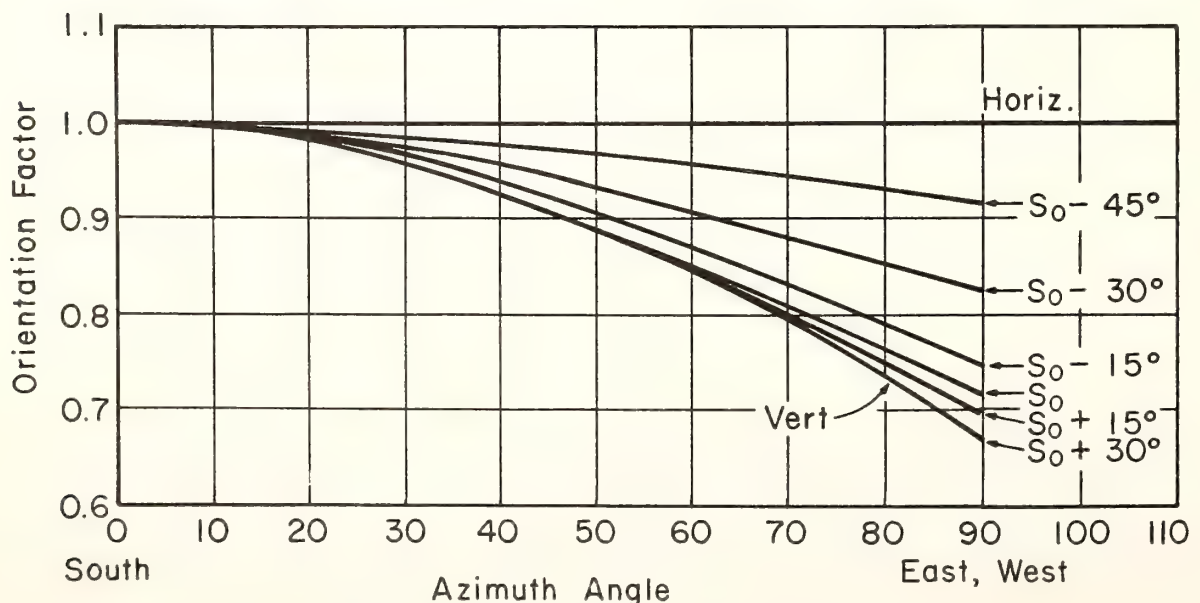


Figure 3-8. Effect of Collector Orientation

CALCULATIONS FOR SYSTEM DESIGN

The monthly average daily solar radiation on a tilted surface that is facing south, \bar{I}_T , (zero azimuth) is the quantity needed to select the collector area of a solar heating system. The quantity, \bar{I}_T , is calculated from the mean daily horizontal radiation, \bar{I}_H , with a conversion factor R that is calculated for a specific location.

The quantities needed to calculate \bar{I}_T are illustrated in Figure 3-9, and Equation (3-1) is to be used.

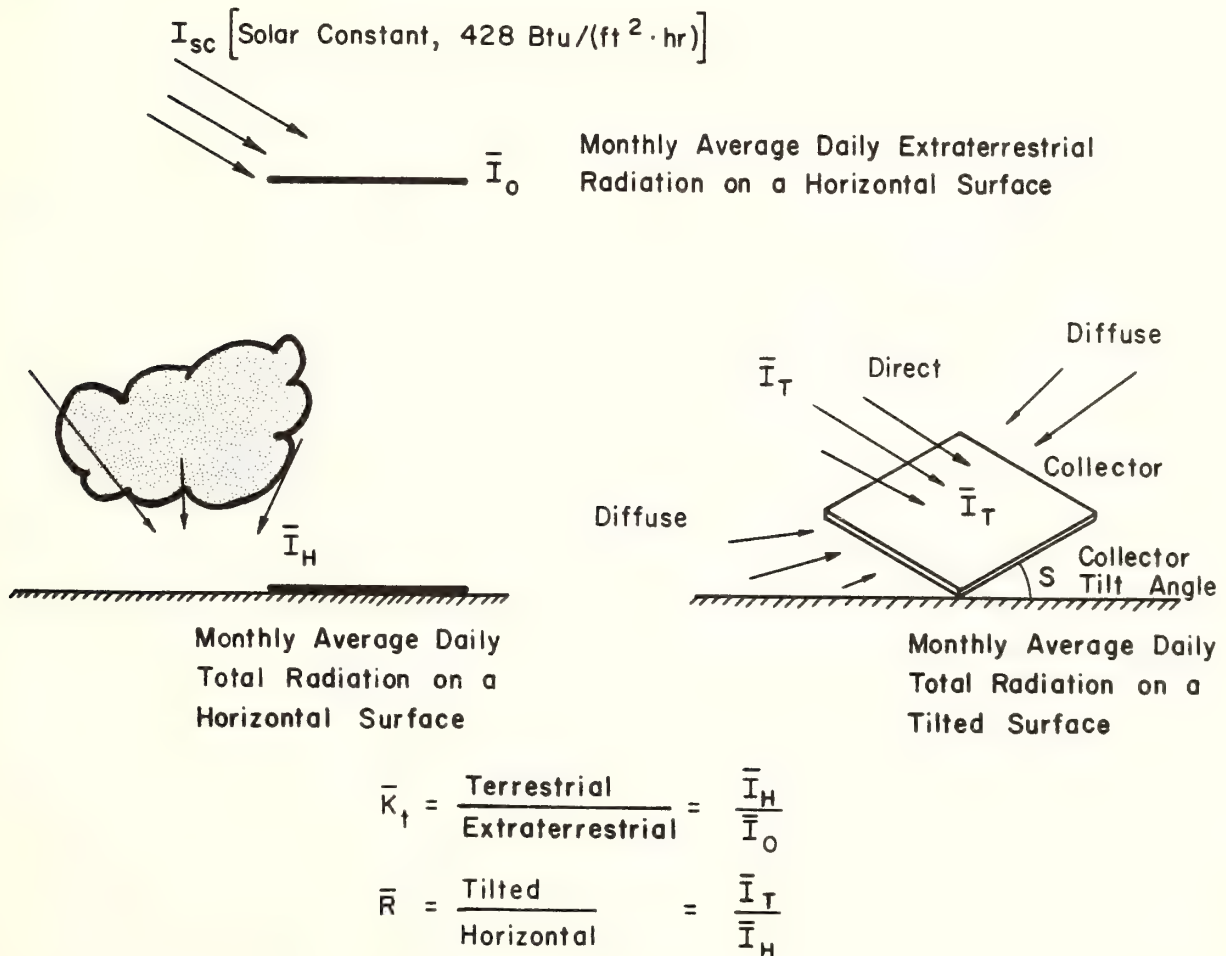


Figure 3-9. Solar Radiation on Horizontal and Tilted Surfaces

$$\bar{I}_T = \bar{R} \cdot \bar{I}_H \quad (3-1)$$

where

\bar{I}_H is determined from Table A3-2 or radiation maps of Figures A3-1 through A3-12

\bar{R} is calculated from Equation (3-2), or interpolated from Tables A3-3 through A3-7, or read from graphs of Figures A3-13 through A3-40.

$$\bar{R} = (1 - \frac{\bar{D}}{\bar{I}_H}) R_d + \frac{\bar{D}}{\bar{I}_H} (\frac{1+\cos s}{2}) + \rho (\frac{1-\cos s}{2}) \quad (3-2)$$

where

s is collector tilt in degrees

ρ is ground reflectance. A value of 0.25 may be representative for green grass, 0.2 for old concrete and crushed rock, and 0.1 for bituminous parking lot surface

$\frac{\bar{D}}{\bar{I}_H}$ is determined from Equation (3-3), and

R_d is determined from Equation (3-5).

$$\frac{\bar{D}}{\bar{I}_H} = 1.3903 - 4.0273 \bar{K}_t + 5.532 \bar{K}_t^2 - 3.100 \bar{K}_t^3, \dots \quad (3-3)$$

and

$$K_t = \frac{\bar{I}_H}{\bar{I}_0} \quad (3-4)$$

$$R_d = \frac{\cos(\phi-s) \cos \delta \sin \omega_s' + \omega_s'' \sin(\phi-s) \sin \delta}{\cos \phi \cos \delta \sin \omega_s + \omega_s^* \sin \phi \sin \delta} \quad (3-5)$$

in which,

ω_s' is the smaller value of [ω_s or $\cos^{-1}(\tan(\phi-s) \tan \delta)$]

ω_s'' is ω_s' expressed in radians

ϕ is latitude angle, degrees

s is collector tilt angle, degrees

δ is declination estimated from Equation (3-6), degrees

ω_s is sunset hour angle calculated by Equation (3-7), degrees

ω_s^* is sunset hour angle expressed in radians

$$\delta = 23.45 \sin \left[360 \left(\frac{284+n}{365} \right) \right] \quad (3.6)$$

in which,

n is the day of the year counted from January 1

$$\omega_s = \cos^{-1} (-\tan \phi \tan \delta) \quad (3-7)$$

The calculation of the radiation on a tilted surface, \bar{I}_T , by Equations (3-2) through (3-7) can be time consuming (and confusing), so it is more convenient to use a table of \bar{R} values and even more convenient to read \bar{R} values from graphs. Because the value of \bar{R} depends on many variables, \bar{R} values not listed in Tables A3-3 through A3-7 must be interpolated from the tabular values (for different \bar{K}_t), latitude, collector tilt, and month of year. Interpolation of \bar{R} values is somewhat easier with the graphs of Figures A3-13 through A3-40. Values of \bar{K}_t for 81 cities are listed for each month in Table A3-2, and for locations that are not listed, estimates need to be made, either from nearby locations or by reading the solar radiation maps for \bar{I}_H and the

appropriate value of \bar{I}_0 from Table A3-1 and solving Equation (3-4). The extraterrestrial radiation may also be calculated from Equation (3-8)

$$\bar{I}_0 = \frac{1}{\Delta t} \int_0^{\Delta t} \frac{24}{\pi} I_{sc} [1 + 0.033 \cos(\frac{360n}{365})] [\cos \phi \cos \delta \sin \omega_s + \omega_s \frac{2\pi}{360} \sin \phi \sin \delta] dt \quad (3-8)$$

EXAMPLE PROBLEMS

EXAMPLE PROBLEM 1

Determine the average daily radiation for the month of January on a collector surface that is tilted at an angle of 51 degrees in Las Vegas, Nevada.

Solution

Step 1. From Table A3-2

Read: $\bar{I}_H = 1035.8 \text{ Btu}/(\text{ft}^2 \cdot \text{day})$

$\bar{K}_t = 0.654$

Latitude = $36^\circ 05' \text{ N}$

Tilt = Latitude + 15°

Step 2. From Table A3-6 (for $\bar{K}_t = 0.6$), and tilt = latitude + 15° ,

Read: Latitude 35° $\bar{R} = 1.76$

Latitude 40° $\bar{R} = 2.02$

Step 3. Calculate for Latitude $36^\circ 05' (36.08^\circ)$

$$\bar{R} = \left(\frac{2.02 - 1.76}{40 - 35} \right) \times (36.08 - 35) + 1.76 = 1.82$$

Step 4. From Table A3-7 (for $\bar{K}_t = 0.7$), and tilt = latitude + 15°

Read: Latitude 35° $\bar{R} = 1.86$

Latitude 40° $\bar{R} = 2.15$

Step 5. Calculate for Latitude 36° 05'

$$\bar{R} = \left(\frac{2.15 - 1.86}{40 - 35} \right) \times (36.08 - 35) + 1.86 = 1.92$$

Step 6. Calculate for $\bar{K}_t = 0.654$

$$\bar{R} = \left(\frac{1.92 - 1.82}{0.7 - 0.6} \right) \times (0.654 - 0.6) + 1.82 = 1.87$$

Step 7. $\bar{I}_T = \bar{R} \cdot \bar{I}_H$

$$= 1.87 \times 1035.8 = \underline{\underline{1937 \text{ Btu}/(\text{ft}^2 \cdot \text{day})}} \quad \text{ANSWER}$$

Alternate Solution to Problem 1

Step 1. From Table A3-2

Read: $\bar{I}_H = 1035.8 \text{ Btu}/(\text{ft}^2 \cdot \text{day})$

$\bar{K}_t = 0.654$

Latitude = 36° 05' N

Step 2. Calculate

$$\text{Latitude} - \text{tilt} = 36^\circ - 51^\circ = -15^\circ$$

$$\text{or, Tilt} = \text{Latitude} + 15^\circ$$

Step 3. From Figure A3-29 (Latitude 35°, Tilt 50°)

Read for $\bar{K}_t = 0.654$, $\bar{R} = 1.81$

Step 4. From Figure A3-30 (Latitude 40°, Tilt 55°)

Read for $\bar{K}_t = 0.654$, $\bar{R} = 2.09$

Step 5. Interpolate for latitude = 36° 05" (36.08°)

$$\bar{R} = \left(\frac{2.09 - 1.81}{40 - 35} \right) \times (36.08 - 35) + 1.81 = 1.87$$

(compare with Step 6, previous solution, 1.87)

Step 6. $\bar{I}_T = \bar{R} \cdot \bar{I}_H$

$$= 1.87 \times 1035.8 = \underline{\underline{1937 \text{ Btu}/(\text{ft}^2 \cdot \text{day})}} \quad \text{ANSWER}$$

EXAMPLE PROBLEM 2

Determine the average daily radiation for the month of January on a collector surface that is tilted at an angle of 45 degrees for a solar heating system in Kansas City, Missouri.

Solution

Step 1. In Table A3-2 note that Kansas City, MO is not listed.

Step 2. Find from Figure A3-1

$$\bar{I}_H = 190 \text{ Langleys/day} = 190 \times 3.69 = 701 \text{ Btu}/(\text{ft}^2 \cdot \text{day})$$

$$\text{Latitude} = 39.5^\circ$$

Step 3. Calculate \bar{K}_t

From Table A3-1

$$\text{Latitude } 35^\circ \quad \bar{I}_O = 1590 \text{ Btu}/(\text{ft}^2 \cdot \text{day})$$

$$\text{Latitude } 40^\circ \quad \bar{I}_O = 1324 \text{ Btu}/(\text{ft}^2 \cdot \text{day})$$

Interpolate for Latitude 39.5°

$$\bar{I}_O = \left(\frac{1324-1590}{40-35} \right) \times (39.5-40) + 1324 = 1351 \text{ Btu}/(\text{ft}^2 \cdot \text{day})$$

$$\bar{K}_t = \frac{\bar{I}_H}{\bar{I}_O} = \frac{701}{1351} = 0.519.$$

Step 4. Calculate (latitude - tilt)

$$\text{Latitude} - \text{tilt} = (39.5 - 45) = -5.5^\circ$$

$$\text{or, Tilt} = \text{Latitude} + 5.5^\circ$$

Step 5. From Figure A3-29 (for latitude 35° , tilt = latitude + 15°)

$$\text{Read: for } \bar{K}_t = 0.519, \bar{R} = 1.67$$

$$\text{From Figure A3-30 (for latitude } 40^\circ, \text{ tilt} = \text{latitude} + 15^\circ) \text{ for } \bar{K}_t = 0.519, \bar{R} = 1.92$$

Interpolate for latitude 39.5°

$$\bar{R} = \left(\frac{1.92-1.67}{40-35} \right) \times (3.95-35) + 1.67 = 1.90$$

Step 6. From Figure A3-22 (for latitude 35° , tilt = latitude)

Read: for $\bar{K}_t = 0.519$, $\bar{R} = 1.57$

From Figure A3-23 (for latitude 40° , tilt = latitude)

for $\bar{K}_t = 0.519$, $\bar{R} = 1.79$

Interpolate for latitude 39.5°

$$\bar{R} = \left(\frac{1.79-1.57}{40-35} \right) \times (39.5-35) + 1.57 = 1.77$$

Step 7. Interpolate for collector tilt = latitude + 5.5°

$$\bar{R} = \left(\frac{1.90-1.77}{15^\circ} \right) \times (5.5^\circ) + 1.77 = 1.82$$

Step 8. Calculate \bar{I}_T

$$\bar{I}_T = \bar{R} \cdot \bar{I}_H$$

$$= (1.82)(701) = \underline{\underline{1276 \text{ Btu}/(\text{ft}^2 \cdot \text{day})}} \quad \text{ANSWER}$$

Alternate Solution to Problem 2

Step 1. Find from Figure A3-1

$$\bar{I}_H = 190 \text{ Langleys/day} = 190 \times 3.69 = 701 \text{ Btu}/(\text{ft}^2 \cdot \text{day})$$

Latitude = 39.5°

Step 2. Calculate \bar{K}_t

From Table A3-1: Latitude 35° $\bar{I}_o = 1590 \text{ Btu}/(\text{ft}^2 \cdot \text{day})$

Latitude 40° $\bar{I}_o = 1324 \text{ Btu}/(\text{ft}^2 \cdot \text{day})$

Interpolate for Latitude 39.5°

$$\bar{I}_o = \left(\frac{1590-1324}{40-35} \right) \times (40-39.5) + 1324 = 1351 \text{ Btu}/(\text{ft}^2 \cdot \text{day})$$

$$\bar{K}_t = \frac{\bar{I}_H}{\bar{I}_o} = \frac{701}{1351} = 0.519$$

Step 3. Calculate \bar{R} from Equations (3-2) through (3-7)

Use $\rho = 0.2$; $n = 15$; $s = 45^\circ$

Result $\bar{R} = 1.83$

(compare with 1.82 from previous method)

Step 4. Calculate \bar{I}_T

$$\bar{I}_T = 1.83 \times 701 = \underline{\underline{1283 \text{ Btu}/(\text{ft}^2 \cdot \text{day})}} \quad \text{ANSWER.}$$

REFERENCES

1. Jessup, E., "A Brief History of the Solar Radiation Program", Report and Recommendations of the Solar Energy Data Workshop, November 29-30, 1973. Report No. NSF-RANN-74-062, NOAA, September 1974.
2. Liu, B. Y. H., and Jordan, R. C., (1977), "Availability of Solar Energy for Flat-Plate Solar Heat Collectors", Chapter V, Applications of Solar Energy for Heating and Cooling of Buildings. ASHRAE GRP 170 edited by Jordan and Liu, ASHRAE, Inc., New York, New York.
3. Klein, S. A., Beckman, W. A., and Duffie, J. A., "A Design Procedure for Solar Heating Systems". Presented by International Solar Energy Society Meeting, Los Angeles, California, July/August 1975.
4. Liu, B. Y. H., and Jordan, R. C., (1963) "A Rational Procedure for Predicting a Long-Term Average Performance of Flat-Plate Collectors", Solar Energy, Volume 7, No. 2.
5. Liu, B. Y. H., and Jordan, R. C., (1960) "The Interrelationship and Characteristic Distribution of Direct, Diffuse, and Total Solar Radiation." Solar Energy, Volume 4, No. 3, pp. 1-19.
6. Duffie, J. A., and Beckman, W. A., (1974) Solar Energy Thermal Processes, John Wiley and Sons, New York, New York.
7. National Bureau of Standards, Intermediate Minimum Property Standards for Solar Heating and Domestic Hot Water Systems. Report No. NBSIR 77-1226, March 1977.

APPENDIX

Table A3-1

\bar{I}_0 , Monthly Average Daily Extraterrestrial Radiation on a Horizontal Surface Btu/(ft²·day)

Location	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
20	2349	2671	3019	3301	3421	3445	3423	3332	3106	2763	2421	2246
25	2103	2474	2891	3266	3463	3524	3485	3329	3013	2588	2192	1995
30	1851	2260	2740	3206	3482	3581	3526	3303	2877	2395	1950	1735
35	1590	2030	2570	3124	3479	3619	3546	3254	2759	2184	1698	1468
40	1324	1788	2380	3019	3454	3637	3545	3183	2600	1950	1438	1194
45	1056	1535	2172	2892	3409	3636	3525	3090	2421	1720	1174	931
50	719	1275	1948	2746	3346	3621	3489	2979	2225	1470	910	669
55	535	1011	1769	2582	3209	3596	3441	2856	2012	1212	651	422
60	299	747	1459	2403	3185	3571	3389	2709	1784	950	405	200

Table A3-2

Radiation and Other Data for 81 Locations in the United States*

		Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
ALASKA													
Annette Is.....	\bar{T}_H	236.2	428.4	883.4	1357.2	1634.7	1638.7	1632.1	1269.4	962	454.6	220.3	152
Lat. 55°02'N.....	K_t	0.427	0.415	0.492	0.507	0.484	0.441	0.454	0.427	0.449	0.347	0.304	0.361
EI. 110 ft.....	t_a	35.8	37.5	39.7	44.4	51.0	56.2	58.6	59.8	54.8	48.2	41.9	37.4
Barrow.....	\bar{T}_H	13.3	143.2	713.3	1491.5	1883	2055.3	1602.2	953.5	428.4	152.4	22.9	-
Lat. 71°20'N.....	K_t	-	0.776	0.773	0.726	0.553	0.533	0.448	0.377	0.315	0.35	-	-
EI. 22 ft.....	t_a	-13.2	-15.9	-12.7	2.1	20.5	35.4	41.6	40.0	31.7	18.6	2.6	-8.6
Bethel.....	\bar{T}_H	142.4	404.8	1052.4	1662.3	1711.8	1698.1	1401.8	938.7	755	430.6	164.9	83
Lat. 60°47'N.....	K_t	0.536	0.557	0.704	0.675	0.519	0.458	0.398	0.336	0.406	0.432	0.399	0.459
EI. 125 ft.....	t_a	9.2	11.6	14.2	29.4	42.7	55.5	56.9	54.8	47.4	33.7	19.0	9.4
Fairbanks.....	\bar{T}_H	66	283.4	860.5	1481.2	1806.2	1970.8	1702.9	1247.6	699.6	323.6	104.1	20.3
Lat. 64°49'N.....	K_t	0.639	0.556	0.674	0.647	0.546	0.529	0.485	0.463	0.419	0.416	0.47	0.458
EI. 436 ft.....	t_a	-7.0	0.3	13.0	32.2	50.5	62.4	63.8	58.3	47.1	29.6	5.5	-6.6
Matanuska.....	\bar{T}_H	119.2	345	-	1327.6	1628.4	1727.6	1526.9	1169	737.3	373.8	142.8	56.4
Lat. 61°30'N.....	K_t	0.513	0.503	-	0.545	0.494	0.466	0.434	0.419	0.401	0.390	0.372	0.364
EI. 180 ft.....	t_a	13.9	21.0	27.4	38.6	50.3	57.6	60.1	58.1	50.2	37.7	22.9	13.9
ALBERTA													
Edmonton.....	\bar{T}_H	331.7	652.4	1165.3	1541.7	1900.4	1914.4	1964.9	1528	1113.3	704.4	413.6	245
Lat. 53°35'N.....	K_t	0.529	0.585	0.624	0.564	0.558	0.514	0.549	0.506	0.506	0.504	0.510	0.492
EI. 2219 ft.....	t_a	10.4	14	26.3	42.9	55.4	61.3	66.6	63.2	54.2	44.1	26.7	14.0
ARKANSAS													
Little Rock.....	\bar{T}_H	704.4	974.2	1335.8	1669.4	1960.1	2091.5	2081.2	1938.7	1640.6	1282.6	913.6	701.1
Lat. 34°44'N.....	K_t	0.424	0.458	0.496	0.513	0.545	0.599	0.566	0.574	0.561	0.552	0.484	0.463
EI. 265 ft.....	t_a	44.6	48.5	56.0	65.8	73.1	76.7	85.1	84.6	78.3	67.9	54.7	46.7
ARIZONA													
Phoenix.....	\bar{T}_H	1126.6	1514.7	1967.1	2388.2	2709.6	2781.5	2450.5	2299.6	2131.3	1688.9	1290	1040.9
Lat. 32°26'N.....	K_t	0.65	0.691	0.716	0.728	0.753	0.745	0.667	0.677	0.722	0.708	0.657	0.652
EI. 1112 ft.....	t_a	54.2	58.8	64.7	72.2	80.8	89.2	94.6	92.5	87.4	75.8	63.6	56.7
Tucson.....	\bar{T}_H	1171.9	1453.8	-	2434.7	-	2601.4	2292.2	2179.7	2122.5	1640.9	1322.1	1132.1
Lat. 32°07'N.....	K_t	0.648	0.646	-	0.738	-	0.698	0.625	0.640	0.710	0.672	0.650	0.679
EI. 2556 ft.....	t_a	53.7	57.3	62.3	69.7	78.0	87.0	90.1	87.4	84.0	73.9	62.5	56.1
CALIFORNIA													
Davis.....	\bar{T}_H	599.2	945	1504	1959	2368.6	2619.2	2565.6	2287.8	1856.8	1288.5	795.6	550.5
Lat. 38°33'N.....	K_t	0.416	0.490	0.591	0.617	0.662	0.697	0.697	0.687	0.664	0.598	0.477	0.421
EI. 51 ft.....	t_a	47.6	52.1	56.8	63.1	69.6	75.7	81	79.4	76.7	67.8	57	48.7
Fresno.....	\bar{T}_H	712.9	1116.6	1652.8	2049.4	2409.2	2641.7	2512.2	2300.7	1897.8	1415.5	906.6	616.6
Lat. 36°46'N.....	K_t	0.462	0.551	0.632	0.638	0.672	0.703	0.682	0.686	0.665	0.635	0.512	0.44
EI. 331 ft.....	t_a	47.3	53.9	59.1	65.6	73.5	80.7	87.5	84.9	78.6	68.7	57.3	48.9
Inyokern.....	\bar{T}_H	1148.7	1554.2	2136.9	2594.8	2925.4	3108.8	2908.8	2759.4	2409.2	1819.2	3170.1	1094.4
Lat. 35°39'N.....	K_t	0.716	0.745	0.803	0.8	0.815	0.830	0.790	0.820	0.834	0.795	0.743	0.742
EI. 2440 ft.....	t_a	47.3	53.9	59.1	65.6	73.5	80.7	87.5	84.9	78.6	68.7	57.3	48.9
Los Angeles, (WBO).....	\bar{T}_H	911.8	1223.6	1640.9	1866.8	2061.2	2259	2428.4	2198.9	1891.5	1362.3	1053.1	877.8
Lat. 34°03'N.....	K_t	0.538	0.568	0.602	0.571	0.573	0.605	0.66	0.648	0.643	0.578	0.548	0.566
EI. 99 ft.....	t_a	57.9	59.2	61.8	64.3	67.6	70.7	75.8	76.1	74.2	69.6	65.4	60.2
Los Angeles, (WBAS).....	\bar{T}_H	930.6	1284.1	1729.5	1948	2196.7	2272.3	2413.6	2155.3	1898.1	1372.7	1082.3	901.1
Lat. 33°56'N.....	K_t	0.547	0.596	0.635	0.595	0.610	0.608	0.657	0.635	0.641	0.574	0.551	0.566
EI. 99 ft.....	t_a	56.2	56.9	59.2	61.4	64.2	66.7	69.6	70.2	69.1	66.1	62.6	58.7
Riverside.....	\bar{T}_H	999.6	1335	1750.5	1943.2	2282.3	2492.6	2443.5	2263.8	1955.3	1509.6	1169	979.7
Lat. 33°57'N.....	K_t	0.589	0.617	0.643	0.594	0.635	0.667	0.665	0.668	0.665	0.639	0.606	0.626
EI. 1020 ft.....	t_a	55.3	57.0	60.6	65.0	69.4	74.0	81.0	81.0	78.5	71.0	63.1	57.2
Santa Maria.....	\bar{T}_H	983.8	1296.3	1805.9	2067.9	2375.6	2599.6	2540.6	2293.3	1965.7	1566.4	1169	943.9
Lat. 34°54'N.....	K_t	0.595	0.613	0.671	0.636	0.661	0.695	0.690	0.678	0.674	0.676	0.624	0.627
EI. 238 ft.....	t_a	54.1	55.3	57.6	59.5	61.2	63.5	65.3	65.7	65.9	64.1	60.8	56.1
COLORADO													
Grand Junction.....	\bar{T}_H	848	1210.7	1622.9	2002.2	2300.3	2645.4	2517.7	2157.2	1957.5	1394.8	969.7	793.4
Lat. 39°07'N.....	K_t	0.597	0.633	0.643	0.632	0.643	0.704	0.690	0.65	0.705	0.654	0.59	0.621
EI. 4849 ft.....	t_a	26.9	35.0	44.6	55.8	66.3	75.7	82.5	79.6	71.4	58.3	42.0	31.4
Grand Lake.....	\bar{T}_H	735	1135.4	1579.3	1876.7	1974.9	2369.7	2103.3	1708.5	1715.8	1212.2	775.6	660.5
Lat. 40°15'N.....	K_t	0.541	0.615	0.637	0.597	0.553	0.63	0.572	0.516	0.626	0.583	0.494	0.542
EI. 8389 ft.....	t_a	18.5	23.1	28.5	39.1	48.7	56.6	62.8	61.5	55.5	45.2	30.3	22.6

*Taken from Applications of Solar Energy for Heating and Cooling of Buildings, ASHRAE [2].

Table A3-2 (continued)

		Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
<u>DISTRICT OF COLUMBIA</u>													
Washington (WBCO).....	\bar{T}_H	632.4	901.5	1255	1600.4	1846.8	2080.8	1929.9	1712.2	1446.1	1083.4	763.5	594.1
Lat. 38°51'N.....	K_t	0.445	0.470	0.496	0.504	0.516	0.553	0.524	0.516	0.520	0.506	0.464	0.460
El. 64 ft.....	t_a	38.4	39.6	48.1	57.5	67.7	76.2	79.9	77.9	72.2	60.9	50.2	40.2
<u>FLORIDA</u>													
Apalachicola.....	\bar{T}_H	1107	1378.2	1654.2	2040.9	2268.6	2195.9	1978.6	1912.9	1703.3	1544.6	1243.2	982.3
Lat. 29°45'N.....	K_t	0.577	0.584	0.576	0.612	0.630	0.594	0.542	0.558	0.559	0.608	0.574	0.543
El. 35 ft.....	t_a	57.3	59.0	62.9	69.5	76.4	81.8	83.1	83.1	80.6	73.2	63.7	58.55
Gainesville.....	\bar{T}_H	1036.9	1324.7	1635	1956.4	1934.7	1960.9	1895.6	1873.8	1615.1	1312.2	1169.7	919.5
Lat. 29°39'N.....	K_t	0.535	0.56	0.568	0.587	0.538	0.531	0.519	0.547	0.529	0.515	0.537	0.508
El. 165 ft.....	t_a	62.1	63.1	67.5	72.8	79.4	83.4	83.8	84.1	82	75.7	67.2	62.4
Miami.....	\bar{T}_H	1292.2	1554.6	1828.8	2020.6	2068.6	1991.5	1992.6	1890.8	1646.8	1436.5	1321	1183.4
Lat. 25°47'N.....	K_t	0.604	0.616	0.612	0.600	0.578	0.545	0.552	0.549	0.525	0.534	0.559	0.588
El. 9 ft.....	t_a	71.6	72.0	73.8	77.0	79.9	82.9	84.1	84.5	83.3	80.2	75.6	72.6
Tampa.....	\bar{T}_H	1223.6	1461.2	1771.9	2016.2	2228	2146.5	1991.9	1845.4	1687.8	1493.3	1328.4	1119.5
Lat. 27°55'N.....	K_t	0.605	0.600	0.606	0.602	0.620	0.583	0.548	0.537	0.546	0.572	0.590	0.589
El. 11 ft.....	t_a	64.2	65.7	68.8	74.3	79.4	83.0	84.0	84.4	82.9	77.2	69.6	65.5
<u>GEORGIA</u>													
Atlanta.....	\bar{T}_H	848	1080.1	1426.9	1807	2618.1	2002.6	2002.9	1898.1	1519.2	1290.8	997.8	751.6
Lat. 33°39'N.....	K_t	0.493	0.496	0.522	0.551	0.561	0.564	0.545	0.559	0.515	0.543	0.510	0.474
El. 976 ft.....	t_a	47.2	49.6	55.9	65.0	73.2	80.9	82.4	81.6	77.4	66.5	54.8	47.7
Griffin.....	\bar{T}_H	889.6	1135.8	1450.9	1923.6	2163.1	2176	2064.9	1961.2	1605.9	1352.4	1073.8	781.5
Lat. 33°15'N.....	K_t	0.513	0.517	0.528	0.586	0.601	0.583	0.562	0.578	0.543	0.565	0.545	0.487
El. 980 ft.....	t_a	48.9	51.0	59.1	66.7	74.6	81.2	83.0	82.2	78.4	68	57.3	49.4
<u>IDAHO</u>													
Boise.....	\bar{T}_H	518.8	884.9	1280.4	1814.4	2189.3	2376.7	2500.3	2149.4	1717.7	1128.4	678.6	456.8
Lat. 43°34'N.....	K_t	0.446	0.533	0.548	0.594	0.619	0.631	0.684	0.660	0.656	0.588	0.494	0.442
El. 2844 ft.....	t_a	29.5	36.5	45.0	53.5	62.1	69.3	79.6	77.2	66.7	56.3	42.3	33.1
<u>ILLINOIS</u>													
Lemont.....	\bar{T}_H	(590)	879	1255.7	1481.5	1866	2041.7	1990.8	1836.9	1469.4	1015.5	(639)	(531)
Lat. 41°40'N.....	K_t	(0.464)	0.496	0.520	0.477	0.525	0.542	0.542	0.559	0.547	0.506	(0.433)	(0.467)
El. 595 ft.....	t_a	28.9	30.3	39.5	49.7	59.2	70.8	75.6	74.3	67.2	57.6	43.0	30.6
<u>INDIANA</u>													
Indianapolis.....	\bar{T}_H	526.2	797.4	1184.1	1481.2	1828	2042	2039.5	1832.1	1513.3	1094.4	662.4	491.1
Lat. 39°44'N.....	K_t	0.380	0.424	0.472	0.47	0.511	0.543	0.554	0.552	0.549	0.520	0.413	0.391
El. 793 ft.....	t_a	31.3	33.9	43.0	54.1	64.9	74.8	79.6	77.4	70.6	59.3	44.2	33.4
<u>KANSAS</u>													
Dodge City.....	\bar{T}_H	953.1	1186.3	1565.7	1975.6	2126.5	2459.8	2400.7	2210.7	1841.7	1421	1065.3	873.8
Lat. 37°46'N.....	K_t	0.639	0.598	0.606	0.618	0.594	0.655	0.652	0.663	0.654	0.650	0.625	0.652
El. 2592 ft.....	t_a	33.8	38.7	46.5	57.7	66.7	77.2	83.8	82.4	73.7	61.7	46.5	36.8
<u>KENTUCKY</u>													
Lexington.....	\bar{T}_H	-	-	-	1834.7	2171.2	-	2246.5	2064.9	1775.6	1315.8	-	681.5
Lat. 38°02'N.....	K_t	-	-	-	0.575	0.606	-	0.610	0.619	0.631	0.604	-	0.513
El. 979 ft.....	t_a	36.5	38.8	47.4	57.8	67.5	76.2	79.8	78.2	72.8	61.2	47.6	38.5
<u>LOUISIANA</u>													
Lake Charles.....	\bar{T}_H	899.2	1145.7	1487.4	1801.8	2080.4	2213.3	1968.6	1910.3	1678.2	1505.5	1122.1	875.6
Lat. 30°13'N.....	K_t	0.473	0.492	0.521	0.542	0.578	0.597	0.538	0.558	0.553	0.597	0.524	0.494
El. 12 ft.....	t_a	55.3	58.7	63.5	70.9	77.4	83.4	84.8	85.0	81.5	73.8	62.6	56.9
<u>MAINE</u>													
Caribou.....	\bar{T}_H	497	861.6	1360.1	1495.9	1779.7	1779.7	1898.1	1675.6	1254.6	793	415.5	398.9
Lat. 46°52'N.....	K_t	0.504	0.579	0.619	0.507	0.509	0.473	0.522	0.527	0.506	0.455	0.352	0.470
El. 628 ft.....	t_a	11.5	12.8	24.4	37.3	51.8	61.6	67.2	65.0	56.2	44.7	31.3	16.8
Portland.....	\bar{T}_H	565.7	874.5	1329.5	1528.4	1923.2	2017.3	2095.6	1799.2	1428.8	1035	591.5	507.7
Lat. 43°39'N.....	K_t	0.482	0.524	0.569	0.500	0.544	0.536	0.572	0.554	0.546	0.539	0.431	0.491
El. 63 ft.....	t_a	23.7	24.5	34.4	44.8	55.4	65.1	71.1	69.7	61.9	51.8	40.3	28.0
<u>MANITOBA</u>													
Winnipeg.....	\bar{T}_H	488.2	835.4	1354.2	1641.3	1904.4	1962	2123.6	1761.2	1190.4	767.5	444.6	345.4
Lat. 49°54'N.....	K_t	0.601	0.636	0.661	0.574	0.550	0.524	0.587	0.567	0.504	0.482	0.436	0.503
El. 786 ft.....	t_a	3.2	7.1	21.3	40.9	55.9	65.3	71.9	69.4	58.6	45.6	25.2	10.1
<u>MASSACHUSETTS</u>													
Blue Hill.....	\bar{T}_H	555.3	797	1143.9	1438	1776.4	1943.9	1881.5	1622.1	1314	941	592.2	482.3
Lat. 42°13'N.....	K_t	0.445	0.458	0.477	0.464	0.501	0.516	0.513	0.495	0.492	0.472	0.406	0.436
El. 629 ft.....	t_a	28.3	28.3	36.9	46.9	58.5	67.2	72.3	70.6	64.2	54.1	43.3	31.5
Boston.....	\bar{T}_H	505.5	738	1067.1	1355	1769	1864	1860.5	1570.1	1267.5	896.7	535.8	442.8
Lat. 42°22'N.....	K_t	0.410	0.426	0.445	0.438	0.499	0.495	0.507	0.480	0.477	0.453	0.372	0.400
El. 29 ft.....	t_a	31.4	31.4	39.9	49.5	60.4	69.8	74.5	73.8	66.8	57.4	46.6	34.9

Table A3-2 (continued)

		Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
<u>MASSACHUSETTS (Contd.)</u>													
East Wareham.....	I _H	504.4	762.4	1132.1	1392.6	1704.8	1958.3	1873.8	1607.4	1363.8	996.7	636.2	521
Lat. 41°46'N.....	K _t	0.398	0.431	0.469	0.449	0.480	0.520	0.511	0.489	0.508	0.496	0.431	0.461
EI. 18 ft.....	t _a	32.2	31.6	39.0	48.3	58.9	67.5	74.1	72.8	65.9	56	46	34.8
<u>MICHIGAN</u>													
East Lansing.....	I _H	425.8	739.1	1086	1249.8	1732.8	1914	1884.5	1627.7	1303.3	891.5	473.1	379.7
Lat. 42°44'N.....	K _t	0.35	0.431	0.456	0.406	0.489	0.508	0.514	0.498	0.493	0.456	0.333	0.349
EI. 856 ft.....	t _a	26.0	26.4	35.7	48.4	59.8	70.3	74.5	72.4	65.0	53.5	40.0	29.0
Sault Ste. Marie.....	I _H	488.6	843.9	1336.5	1559.4	1962.3	2064.2	2149.4	1767.9	1207	809.2	392.2	359.8
Lat. 46°28'N.....	K _t	0.490	0.560	0.606	0.526	0.560	0.549	0.590	0.554	0.481	0.457	0.323	0.408
EI. 724 ft.....	t _a	16.3	16.2	25.6	39.5	52.1	61.6	67.3	66.0	57.9	46.8	33.4	21.9
<u>MINNESOTA</u>													
St. Cloud.....	I _H	632.8	976.7	1383	1598.1	1859.4	2003.3	2087.8	1828.4	1369.4	890.4	545.4	463.1
Lat. 45°35'N.....	K _t	0.595	0.629	0.614	0.534	0.530	0.533	0.573	0.570	0.539	0.490	0.435	0.504
EI. 1034 ft.....	t _a	13.6	16.9	29.8	46.2	58.8	68.5	74.4	71.9	62.5	50.2	32.1	18.3
<u>MISSOURI</u>													
Columbia.....	I _H	651.3	941.3	1315.8	1631.3	1999.6	2129.1	2148.7	1953.1	1689.6	1202.6	839.5	590.4
Lat. 38°58'N.....	K _t	0.458	0.492	0.520	0.514	0.559	0.566	0.585	0.588	0.606	0.562	0.510	0.457
EI. 785 ft.....	t _a	32.5	36.5	45.9	57.7	66.7	75.9	81.1	79.4	71.9	61.4	46.1	35.8
<u>MONTANA</u>													
Glasgow.....	I _H	572.7	965.7	1437.6	1741.3	2127.3	2261.6	2414.7	1984.5	1531	997	574.9	428.4
Lat. 48°13'N.....	K _t	0.621	0.678	0.672	0.597	0.611	0.602	0.666	0.630	0.629	0.593	0.516	0.548
EI. 2277 ft.....	t _a	13.3	17.3	31.1	47.8	59.3	67.3	76	73.2	61.2	49.2	31.0	18.6
Great Falls.....	I _H	524	869.4	1369.7	1621.4	1970.8	2179.3	2383	1986.3	1536.5	984.9	575.3	420.7
Lat. 47°29'N.....	K _t	0.552	0.596	0.631	0.551	0.565	0.580	0.656	0.627	0.626	0.574	0.503	0.518
EI. 3664 ft.....	t _a	25.4	27.6	35.6	47.7	57.5	64.3	73.8	71.3	60.6	51.4	38.0	29.1
<u>NEBRASKA</u>													
Lincoln.....	I _H	712.5	955.7	1299.6	1587.8	1856.1	2040.6	2011.4	1902.6	1543.5	1215.8	773.4	643.2
Lat. 40°51'N.....	K _t	0.542	0.528	0.532	0.507	0.522	0.542	0.547	0.577	0.568	0.596	0.508	0.545
EI. 1189 ft.....	t _a	27.8	32.1	42.4	55.8	65.8	76.0	82.6	80.2	71.5	59.9	43.2	31.8
<u>NEVADA</u>													
Ely.....	I _H	871.6	1255	1749.8	2103.3	2322.1	2649	2417	2307.7	1935	1473	1078.6	814.8
Lat. 39°17'N.....	K _t	0.618	0.660	0.692	0.664	0.649	0.704	0.656	0.695	0.696	0.691	0.658	0.64
EI. 6262 ft.....	t _a	27.3	32.1	39.5	48.3	57.0	65.4	74.5	72.3	63.7	52.1	39.9	31.1
Las Vegas.....	I _H	1035.8	1438	1926.5	2322.8	2629.2	2799.2	2524	2342	2062	1602.6	1190	964.2
Lat. 36°05'N.....	K _t	0.654	0.697	0.728	0.719	0.732	0.746	0.685	0.697	0.716	0.704	0.657	0.668
EI. 2162 ft.....	t _a	47.5	53.9	60.3	69.5	78.3	88.2	95.0	92.9	85.4	71.7	57.8	50.2
<u>NEW JERSEY</u>													
Seabrook.....	I _H	591.9	854.2	1195.6	1518.8	1800.7	1964.6	1949.8	1715	1445.7	1071.9	721.8	522.5
Lat. 39°30'N.....	K _t	0.426	0.453	0.476	0.481	0.504	0.522	0.530	0.517	0.524	0.508	0.449	0.416
EI. 100 ft.....	t _a	39.5	37.6	43.9	54.7	64.9	74.1	79.8	77.7	69.7	61.2	48.5	39.3
<u>NEW MEXICO</u>													
Albuquerque.....	I _H	1150.9	1453.9	1925.4	2343.5	2560.9	2757.5	2561.2	2387.8	2120.3	1639.8	1274.2	1051.6
Lat. 35°03'N.....	K _t	0.704	0.691	0.719	0.722	0.713	0.737	0.695	0.708	0.728	0.711	0.684	0.704
EI. 5314 ft.....	t _a	37.3	43.3	50.1	59.6	69.4	79.1	82.8	80.6	73.6	62.1	47.8	39.4
<u>NEW YORK</u>													
Ithaca.....	I _H	434.3	755	1074.9	1322.9	1779.3	2025.8	2031.3	1736.9	1320.3	918.4	466.4	370.8
Lat. 42°27'N.....	K _t	0.351	0.435	0.45	0.428	0.502	0.538	0.554	0.530	0.497	0.465	0.324	0.337
EI. 950 ft.....	t _a	27.2	26.5	36	48.4	59.6	68.9	73.9	71.9	64.2	53.6	41.5	29.6
New York.....	I _H	539.5	790.8	1180.4	1426.2	1738.4	1994.1	1938.7	1605.9	1349.4	977.8	598.1	476
Lat. 40°46'N.....	K _t	0.406	0.435	0.480	0.455	0.488	0.53	0.528	0.486	0.500	0.475	0.397	0.403
EI. 52 ft.....	t _a	35.0	34.9	43.1	52.3	63.3	72.2	76.9	75.3	69.5	59.3	48.3	37.7
Sayville.....	I _H	602.9	936.2	1259.4	1560.5	1857.2	2123.2	2040.9	1734.7	1446.8	1087.4	697.8	533.9
Lat. 40°30'N.....	K _t	0.453	0.511	0.510	0.498	0.522	0.564	0.555	0.525	0.530	0.527	0.450	0.447
EI. 20 ft.....	t _a	35	34.9	43.1	52.3	63.3	72.2	76.9	75.3	69.5	59.3	48.3	37.7
Schenectady.....	I _H	488.2	753.5	1026.6	1272.3	1553.1	1687.8	1662.3	1494.8	1124.7	820.6	436.2	356.8
Lat. 42°50'N.....	K _t	0.406	0.441	0.433	0.413	0.438	0.448	0.454	0.458	0.426	0.420	0.309	0.331
EI. 217 ft.....	t _a	24.7	24.6	34.9	48.3	61.7	70.8	76.9	73.7	64.6	53.1	40.1	28.0
Upton.....	I _H	583	872.7	1280.4	1609.9	1891.5	2159	2044.6	1789.6	1472.7	1102.6	686.7	551.3
Lat. 40°52'N.....	K _t	0.444	0.483	0.522	0.514	0.532	0.574	0.557	0.542	0.542	0.538	0.448	0.467
EI. 75 ft.....	t _a	35.0	34.9	43.1	52.3	63.3	72.2	76.9	75.3	69.5	59.3	48.3	37.7
<u>NORTH CAROLINA</u>													
Greensboro.....	I _H	743.9	1031.7	1323.2	1755.3	1988.5	2111.4	2033.9	1810.3	1517.3	1202.6	908.1	690.8
Lat. 36°05'N.....	K _t	0.469	0.499	0.499	0.543	0.554	0.563	0.552	0.538	0.527	0.531	0.501	0.479
EI. 891 ft.....	t _a	42.0	44.2	51.7	60.8	69.9	78.0	80.2	78.9	73.9	62.7	51.5	43.2

Table A3-2 (continued)

		Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
<u>NORTH CAROLINA (Contd.)</u>													
Hatteras.....	\bar{T}_H	891.9	1184.1	1590.4	2128	2376.4	2438	2334.3	2085.6	1758.3	1337.6	1053.5	798.1
Lat. 35°13'N.....	K_t	0.546	0.563	0.593	0.655	0.661	0.652	0.634	0.619	0.605	0.58	0.566	0.535
El. 7 ft.....	t_a	49.9	49.5	54.7	61.5	69.9	77.2	80.0	79.8	76.7	67.9	59.1	51.3
<u>NORTH DAKOTA</u>													
Bismarck.....	\bar{T}_H	587.4	934.3	1328.4	1668.2	2056.1	2173.8	2305.5	1929.1	1441.3	1018.1	600.4	464.2
Lat. 46°47'N.....	K_t	0.594	0.628	0.605	0.565	0.588	0.579	0.634	0.606	0.581	0.584	0.510	0.547
El. 1660 ft.....	t_a	12.4	15.9	29.7	46.6	48.6	67.9	76.1	73.5	61.6	49.6	31.4	18.4
<u>OHIO</u>													
Cleveland.....	\bar{T}_H	466.8	681.9	1207	1443.9	1928.4	2102.6	2094.4	1840.6	1410.3	997	526.6	427.3
Lat. 41°24'N.....	K_t	0.361	0.383	0.497	0.464	0.543	0.559	0.571	0.559	0.524	0.491	0.351	0.371
El. 805 ft.....	t_a	30.8	30.9	39.4	50.2	62.4	72.7	77.0	75.1	68.5	57.4	44.0	32.8
Columbus.....	\bar{T}_H	486.3	746.5	1112.5	1480.8	1839.1	(2111)	2041.3	1572.7	1189.3	919.5	479	430.2
Lat. 40°00'N.....	K_t	0.356	0.401	0.447	0.470	0.515	(0.561)	0.555	0.475	0.433	0.441	0.302	0.351
El. 833 ft.....	t_a	32.1	33.7	42.7	53.5	64.4	74.2	78	75.9	70.1	58	44.5	34.0
<u>OKLAHOMA</u>													
Oklahoma City.....	\bar{T}_H	938	1192.6	1534.3	1849.4	2005.1	2355	2273.8	2211	1819.2	1409.6	1085.6	897.4
Lat. 35°24'N.....	K_t	0.580	0.571	0.576	0.570	0.558	0.629	0.618	0.565	0.628	0.614	0.588	0.608
El. 1304 ft.....	t_a	40.1	45.0	53.2	63.6	71.2	80.6	85.5	85.4	77.4	66.5	52.2	43.1
Stillwater.....	\bar{T}_H	763.8	1081.5	1463.8	1702.6	1879.3	2235.8	2224.3	2039.1	1724.3	1314	991.5	783
Lat. 36°09'N.....	K_t	0.484	0.527	0.555	0.528	0.523	0.596	0.604	0.607	0.599	0.581	0.548	0.544
El. 910 ft.....	t_a	41.2	45.6	53.8	64.2	71.6	81.1	85.9	85.9	77.5	67.6	52.6	43.9
<u>ONTARIO</u>													
Ottawa.....	\bar{T}_H	539.1	852.4	1250.5	1506.6	1857.2	2084.5	2045.4	1752.4	1326.6	826.9	458.7	408.5
Lat. 45°20'N.....	K_t	0.499	0.540	0.554	0.502	0.529	0.554	0.560	0.546	0.521	0.450	0.359	0.436
El. 339 ft.....	t_a	14.6	15.6	27.7	43.3	57.5	67.5	71.9	69.8	61.5	48.9	35	19.6
Toronto.....	\bar{T}_H	451.3	674.5	1088.9	1388.2	1785.2	1941.7	1968.6	1622.5	1284.1	835	458.3	352.8
Lat. 43°41'N.....	K_t	0.388	0.406	0.467	0.455	0.506	0.516	0.539	0.500	0.493	0.438	0.336	0.346
El. 379 ft.....	t_a	26.5	26.0	34.2	46.3	58	68.4	73.8	71.8	64.3	52.6	40.9	30.2
<u>OREGON</u>													
Astoria.....	\bar{T}_H	338.4	607	1008.5	1401.5	1838.7	1753.5	2007.7	1721	1322.5	780.4	413.6	295.2
Lat. 46°12'N.....	K_t	0.330	0.397	0.454	0.471	0.524	0.466	0.551	0.538	0.526	0.435	0.336	0.332
El. 8 ft.....	t_a	41.3	44.7	46.9	51.3	55.0	59.3	62.6	63.6	62.2	55.7	48.5	43.9
Medford.....	\bar{T}_H	435.4	804.4	1259.8	1807.4	2216.2	2440.5	2607.4	2261.6	1672.3	1043.5	558.7	346.5
Lat. 42°23'N.....	K_t	0.353	0.464	0.527	0.584	0.625	0.648	0.710	0.689	0.628	0.526	0.384	0.313
El. 1329 ft.....	t_a	39.4	45.4	50.8	56.3	63.1	69.4	76.9	76.4	69.6	58.7	47.1	40.5
<u>PENNSYLVANIA</u>													
State College.....	\bar{T}_H	501.8	749.1	1106.6	1399.2	1754.6	2027.6	1968.2	1690	1336.1	1017	580.1	4443.9
Lat. 40°48'N.....	K_t	0.381	0.413	0.451	0.448	0.493	0.539	0.536	0.512	0.492	0.496	0.379	0.376
El. 1175 ft.....	t_a	31.3	31.4	39.8	51.3	63.4	71.8	75.8	73.4	66.1	55.6	43.2	32.6
<u>RHODE ISLAND</u>													
Newport.....	\bar{T}_H	565.7	856.4	1231.7	1484.8	1849	2019.2	1942.8	1687.1	1411.4	1035.4	656.1	527.7
Lat. 41°29'N.....	K_t	0.438	0.482	0.507	0.477	0.520	0.536	0.529	0.513	0.524	0.512	0.44	0.460
El. 60 ft.....	t_a	29.5	32.0	39.6	48.2	58.6	67.0	73.2	72.3	66.7	56.2	46.5	34.4
<u>SOUTH CAROLINA</u>													
Charleston.....	\bar{T}_H	946.1	1152.8	1352.4	1918.8	2063.4	2113.3	1649.4	1933.6	1557.2	1332.1	1073.8	952
Lat. 32°54'N.....	K_t	0.541	0.521	0.491	0.584	0.574	0.567	0.454	0.569	0.525	0.554	0.539	0.586
El. 46 ft.....	t_a	53.6	55.2	60.6	67.8	74.8	80.9	82.9	82.3	79.1	69.8	59.8	54.0
<u>SOUTH DAKOTA</u>													
Rapid City.....	\bar{T}_H	687.8	1032.5	1503.7	1807	2028	2193.7	2235.8	2019.9	1628	1179.3	763.1	590.4
Lat. 44°09'N.....	K_t	0.601	0.627	0.649	0.594	0.574	0.583	0.612	0.622	0.628	0.624	0.566	0.588
El. 3218 ft.....	t_a	24.7	27.4	34.7	48.2	58.3	67.3	76.3	75.0	64.7	52.9	38.7	29.2
<u>TEXAS</u>													
Brownsville.....	\bar{T}_H	1105.9	1262.7	1505.9	1714	2092.2	2288.5	2345	2124	1774.9	1536.5	1104.8	982.3
Lat. 25°55'N.....	K_t	0.517	0.500	0.505	0.509	0.584	0.627	0.650	0.617	0.566	0.570	0.468	0.488
El. 20 ft.....	t_a	63.3	66.7	70.7	76.2	81.4	85.1	86.5	86.9	84.1	78.9	70.7	65.2
El Paso.....	\bar{T}_H	1247.6	1612.9	2048.7	2447.2	2673	2731	2391.1	2350.5	2077.5	1704.8	1324.7	1051.6
Lat. 31°48'N.....	K_t	0.686	0.714	0.730	0.741	0.743	0.733	0.652	0.669	0.693	0.695	0.647	0.626
El. 3916 ft.....	t_a	47.1	53.1	58.7	67.3	75.7	84.2	84.9	83.4	78.5	69.0	56.0	48.5
Fort Worth.....	\bar{T}_H	936.2	1198.5	1597.8	1829.1	2105.1	2437.6	2293.3	2216.6	1880.8	1476	1147.6	913.6
Lat. 32°50'N.....	K_t	0.530	0.541	0.577	0.556	0.585	0.654	0.624	0.653	0.634	0.612	0.576	0.563
El. 544 ft.....	t_a	48.1	52.3	59.8	68.8	75.9	84.0	87.7	88.6	81.3	71.5	58.8	50.8

Table A3-2 (continued)

		Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
<u>TEXAS (Contd.)</u>													
Midland.....	\bar{I}_H	1066.4	1345.7	1784.8	2036.1	2301.1	2317.7	2301.8	2193	1921.8	1470.8	1244.3	1023.2
Lat. 31°56'N.....	K_t	0.587	0.596	0.638	0.617	0.639	0.622	0.628	0.643	0.642	0.600	0.609	0.611
E1. 2854 ft.....	t_a	47.9	52.8	60.0	68.8	77.2	83.9	85.7	85.0	78.9	70.3	56.6	49.1
San Antonio.....	\bar{I}_H	1045	1299.2	1560.1	1664.6	2024.7	2250*	2364.2	2185.2	1844.6	1487.4	1104.4	954.6
Lat. 29°32'N.....	K_t	0.541	0.550	0.542	0.500	0.563	0.62	0.647	0.637	0.603	0.584	0.507	0.528
E1. 794 ft.....	t_a	53.7	58.4	65.0	72.2	79.2	85.0	87.4	87.8	82.6	74.7	63.3	56.5
<u>TENNESSEE</u>													
Nashville.....	\bar{I}_H	589.7	907	1246.8	1662.3	1997	2149.4	2079.7	1862.7	1600.7	1223.6	823.2	614.4
Lat. 36°07'N.....	K_t	0.373	0.440	0.472	0.514	0.556	0.573	0.565	0.554	0.556	0.540	0.454	0.426
E1. 605 ft.....	t_a	42.6	45.1	52.9	63.0	71.4	80.1	83.2	81.9	76.6	65.4	52.3	44.3
Oak Ridge.....	\bar{I}_H	604	895.9	1241.7	1689.6	1942.8	2066.4	1972.3	1795.6	1559.8	1194.8	796.3	610
Lat. 36°01'N.....	K_t	0.382	0.435	0.471	0.524	0.541	0.551	0.536	0.534	0.542	0.527	0.438	0.422
E1. 905 ft.....	t_a	41.9	44.2	51.7	61.4	69.8	77.8	80.2	78.8	74.5	62.7	50.4	42.5
<u>UTAH</u>													
Salt Lake City.....	\bar{I}_H	622.1	986	1301.1	1813.3	-	-	-	-	1689.3	1250.2	-	552.8
Lat. 40°46'N.....	K_t	0.468	0.909	0.529	0.579	-	-	-	-	0.621	0.610	-	0.467
E1. 4227 ft.....	t_a	29.4	36.2	44.4	53.9	63.1	71.7	81.3	79.0	68.7	57.0	42.5	34.0
<u>WASHINGTON</u>													
Seattle-Tacoma.....	\bar{I}_H	282.6	520.6	992.2	1507	1881.5	1909.9	2110.7	1688.5	1211.8	702.2	386.3	239.5
Lat. 47°27'N.....	K_t	0.296	0.355	0.456	0.510	0.538	0.508	0.581	0.533	0.492	0.407	0.336	0.292
E1. 386 ft.....	t_a	42.1	45.0	48.9	54.1	59.8	64.4	68.4	67.9	63.3	56.3	48.4	44.4
Seattle.....	\bar{I}_H	252	471.6	917.3	1375.6	1664.9	1724	1805.1	1617	1129.1	638	325.5	218.1
Lat. 47°36'N.....	K_t	0.266	0.324	0.423	0.468	0.477	0.459	0.498	0.511	0.459	0.372	0.284	0.269
E1. 14 ft.....	t_a	38.9	42.9	46.9	51.9	58.1	62.8	67.2	66.7	61.6	54.0	45.7	41.5
Spokane.....	\bar{I}_H	446.1	837.6	1200	1864.6	2104.4	2226.5	2479.7	2076	1511	844.6	486.3	279
Lat. 47°40'N.....	K_t	0.478	0.579	0.556	0.602	0.603	0.593	0.684	0.656	0.616	0.494	0.428	0.345
E1. 1968 ft.....	t_a	26.5	31.7	40.5	49.2	57.9	64.6	73.4	71.7	62.7	51.5	37.4	30.5
<u>WISCONSIN</u>													
Madison.....	\bar{I}_H	564.6	812.2	1232.1	1455.3	1745.4	2031.7	2046.5	1740.2	1443.9	993	555.7	495.9
Lat. 43°08'N.....	K_t	0.40	0.478	0.522	0.474	0.493	0.540	0.559	0.534	0.549	0.510	0.396	0.467
E1. 866 ft.....	t_a	21.8	24.6	35.3	49.0	61.0	70.9	76.8	74.4	65.6	53.7	37.8	25.4
<u>WYOMING</u>													
Lander.....	\bar{I}_H	786.3	1146.1	1638	1988.5	2114	2492.2	2438.4	2120.6	1712.9	1301.8	837.3	694.8
Lat. 42°48'N.....	K_t	0.65	0.672	0.691	0.647	0.597	0.662	0.665	0.649	0.647	0.666	0.589	0.643
E1. 5370 ft.....	t_a	20.2	26.3	34.7	45.5	56.0	65.4	74.6	72.5	61.4	48.3	33.4	23.8

*Original values incorrect. Values estimated from insolation maps.

Table A3-3. Ratio of Monthly Average Daily Radiation on a Tilted Surface to that on a Horizontal Surface [Ref. 7]

\bar{R} for $\bar{K}_t = .30$

Latitude	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Tilt = Latitude - 15°												
25	1.09	1.06	1.03	1.00	.98	.98	.98	.99	1.02	1.05	1.08	1.09
30	1.15	1.10	1.05	1.01	.98	.97	.97	.99	1.03	1.08	1.13	1.16
35	1.23	1.15	1.07	1.01	.97	.96	.96	1.00	1.05	1.12	1.20	1.25
40	1.34	1.22	1.11	1.02	.97	.95	.96	1.00	1.07	1.18	1.30	1.38
45	1.51	1.31	1.15	1.03	.97	.94	.95	1.00	1.10	1.25	1.45	1.58
50	1.77	1.44	1.21	1.05	.97	.93	.95	1.01	1.13	1.35	1.67	1.91
55	2.24	1.65	1.29	1.07	.96	.93	.94	1.02	1.18	1.50	2.04	2.53
Tilt = Latitude												
25	1.17	1.11	1.04	.97	.93	.91	.92	.95	1.01	1.08	1.16	1.19
30	1.24	1.15	1.05	.97	.92	.90	.91	.95	1.02	1.11	1.21	1.27
35	1.33	1.20	1.08	.97	.91	.89	.90	.95	1.03	1.16	1.29	1.38
40	1.46	1.27	1.11	.98	.90	.87	.89	.94	1.05	1.21	1.41	1.53
45	1.65	1.37	1.15	.99	.90	.86	.88	.94	1.08	1.29	1.57	1.76
50	1.96	1.52	1.21	1.00	.89	.85	.87	.95	1.11	1.40	1.82	2.14
55	2.51	1.75	1.29	1.01	.89	.84	.86	.95	1.16	1.56	2.25	2.88
Tilt = Latitude + 15°												
25	1.21	1.11	1.00	.91	.84	.82	.83	.88	.96	1.07	1.18	1.24
30	1.28	1.15	1.01	.90	.83	.80	.81	.87	.96	1.10	1.24	1.32
35	1.37	1.20	1.03	.90	.82	.79	.80	.86	.97	1.14	1.32	1.43
40	1.51	1.27	1.06	.90	.81	.77	.79	.86	.99	1.19	1.44	1.60
45	1.71	1.37	1.10	.90	.80	.76	.77	.85	1.01	1.27	1.61	1.84
50	2.04	1.52	1.15	.91	.79	.74	.76	.85	1.04	1.38	1.88	2.26
55	2.63	1.76	1.23	.92	.78	.73	.75	.85	1.08	1.54	2.33	3.05
Tilt = Vertical												
25	.94	.78	.62	.48	.42	.40	.41	.45	.56	.73	.90	.99
30	1.04	.85	.67	.52	.44	.42	.43	.48	.60	.79	.99	1.10
35	1.17	.94	.72	.55	.47	.44	.45	.51	.65	.86	1.10	1.24
40	1.33	1.04	.78	.59	.50	.47	.48	.55	.70	.95	1.25	1.44
45	1.57	1.18	.86	.64	.53	.49	.51	.59	.76	1.06	1.45	1.72
50	1.93	1.36	.95	.68	.56	.52	.54	.63	.82	1.20	1.75	2.17
55	2.55	1.62	1.06	.74	.60	.55	.57	.67	.91	1.40	2.24	3.00

Table A3-4. Ratio of Monthly Average Daily Radiation on a Tilted Surface to that on a Horizontal Surface [Ref. 7]

 \bar{R} for $\bar{K}_t = .40$

Latitude	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Tilt = Latitude -15°												
25	1.11	1.08	1.04	1.01	.98	.97	.98	1.00	1.03	1.07	1.10	1.13
30	1.20	1.13	1.07	1.01	.98	.96	.97	1.00	1.05	1.11	1.18	1.22
35	1.31	1.20	1.11	1.03	.97	.95	.96	1.00	1.07	1.17	1.28	1.34
40	1.46	1.30	1.15	1.04	.97	.94	.96	1.01	1.10	1.25	1.41	1.52
45	1.69	1.43	1.21	1.06	.97	.94	.95	1.02	1.15	1.35	1.61	1.79
50	2.04	1.61	1.30	1.09	.98	.94	.95	1.04	1.20	1.49	1.90	2.22
55	2.68	1.89	1.41	1.12	.98	.93	.95	1.06	1.27	1.70	2.41	3.06
Tilt = Latitude												
25	1.24	1.15	1.06	.98	.92	.90	.91	.95	1.03	1.12	1.22	1.27
30	1.34	1.21	1.09	.98	.91	.88	.90	.95	1.04	1.17	1.30	1.38
35	1.46	1.29	1.13	.99	.91	.87	.89	.95	1.07	1.23	1.41	1.52
40	1.64	1.39	1.17	1.00	.90	.86	.88	.96	1.10	1.31	1.57	1.73
45	1.90	1.53	1.23	1.02	.90	.86	.88	.96	1.14	1.42	1.79	2.04
50	2.32	1.74	1.32	1.04	.90	.85	.87	.98	1.19	1.58	2.13	2.56
55	3.05	2.04	1.43	1.07	.90	.84	.87	.99	1.27	1.80	2.71	3.54
Tilt = Latitude $+15^\circ$												
25	1.31	1.17	1.03	.91	.82	.79	.80	.87	.98	1.12	1.27	1.35
30	1.41	1.23	1.06	.91	.81	.77	.79	.86	.99	1.17	1.36	1.46
35	1.54	1.31	1.09	.91	.80	.76	.78	.86	1.01	1.23	1.47	1.62
40	1.73	1.41	1.13	.92	.80	.75	.77	.86	1.04	1.31	1.64	1.84
45	2.01	1.56	1.19	.93	.79	.74	.76	.87	1.08	1.42	1.87	2.18
50	2.45	1.77	1.27	.95	.79	.73	.76	.88	1.12	1.58	2.23	2.74
55	3.24	2.08	1.39	.98	.79	.72	.75	.89	1.19	1.81	2.85	3.80
Tilt = Vertical												
25	1.05	.84	.63	.44	.36	.34	.35	.40	.54	.77	.99	1.12
30	1.18	.94	.69	.49	.39	.36	.37	.44	.60	.85	1.11	1.26
35	1.35	1.05	.76	.54	.43	.39	.41	.49	.66	.95	1.26	1.45
40	1.57	1.18	.84	.59	.47	.42	.44	.53	.73	1.06	1.46	1.71
45	1.88	1.36	.94	.65	.51	.46	.48	.58	.81	1.21	1.73	2.08
50	2.36	1.60	1.06	.71	.55	.50	.52	.63	.90	1.39	2.12	2.68
55	3.18	1.95	1.21	.78	.60	.54	.56	.69	1.00	1.66	2.76	3.78

Table A3-5. Ratio of Monthly Average Daily Radiation on a Tilted Surface to that on a Horizontal Surface [Ref. 7]

 \bar{R} for $\bar{K}_t = .50$

Latitude	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Tilt = Latitude -15°												
25	1.14	1.09	1.05	1.01	.98	.97	.97	1.00	1.03	1.08	1.12	1.15
30	1.23	1.16	1.08	1.02	.97	.96	.96	1.00	1.06	1.13	1.21	1.26
35	1.37	1.24	1.13	1.03	.97	.95	.96	1.01	1.09	1.20	1.33	1.41
40	1.55	1.36	1.19	1.05	.97	.94	.96	1.02	1.13	1.30	1.49	1.62
45	1.82	1.51	1.26	.98	.98	.94	.96	1.03	1.18	1.42	1.72	1.93
50	2.24	1.73	1.36	1.12	.99	.94	.96	1.06	1.25	1.59	2.08	2.45
55	2.99	2.06	1.50	1.16	1.00	.94	.96	1.08	1.34	1.83	2.67	3.44
Tilt = Latitude												
25	1.29	1.19	1.08	.98	.91	.88	.90	.95	1.04	1.15	1.26	1.32
30	1.40	1.26	1.11	.99	.91	.87	.89	.95	1.06	1.21	1.36	1.45
35	1.56	1.35	1.16	1.00	.90	.86	.88	.96	1.09	1.28	1.50	1.63
40	1.77	1.48	1.22	1.02	.90	.86	.88	.97	1.13	1.38	1.68	1.87
45	2.08	1.65	1.30	1.04	.90	.85	.87	.98	1.18	1.52	1.95	2.25
50	2.57	1.89	1.40	1.08	.91	.85	.87	1.00	1.25	1.70	2.36	2.86
55	3.44	2.26	1.54	1.12	.92	.85	.88	1.02	1.34	1.97	3.04	4.02
Tilt = Latitude + 15°												
25	1.38	1.22	1.05	.91	.81	.77	.79	.86	.99	1.16	1.33	1.43
30	1.50	1.29	1.09	.91	.80	.76	.78	.86	1.01	1.22	1.44	1.57
35	1.66	1.39	1.13	.92	.80	.75	.77	.86	1.04	1.30	1.58	1.75
40	1.89	1.52	1.19	.94	.79	.74	.76	.87	1.08	1.40	1.78	2.02
45	2.22	1.69	1.26	.96	.79	.73	.76	.88	1.12	1.53	2.06	2.43
50	2.75	1.94	1.36	.98	.79	.73	.76	.89	1.19	1.72	2.49	3.09
55	3.68	2.32	1.50	1.02	.80	.72	.75	.91	1.27	1.99	3.22	4.34
Tilt = Vertical												
25	1.13	.89	.63	.42	.32	.29	.30	.37	.53	.80	1.06	1.21
30	1.29	1.00	.71	.47	.35	.32	.33	.41	.60	.89	1.20	1.38
35	1.48	1.13	.79	.53	.40	.35	.37	.47	.67	1.01	1.38	1.60
40	1.74	1.29	.89	.59	.44	.39	.41	.52	.75	1.14	1.61	1.91
45	2.11	1.50	1.00	.66	.49	.44	.46	.58	.84	1.31	1.92	2.34
50	2.67	1.78	1.14	.73	.54	.48	.51	.64	.95	1.54	2.39	3.04
55	3.64	2.19	1.32	.81	.60	.53	.56	.71	1.08	1.84	3.15	4.34

Table A3-6. Ratio of Monthly Average Daily Radiation on a Tilted Surface to that on a Horizontal Surface [Ref. 7]

 \bar{R} for $K_t = .60$

Latitude	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Tilt = Latitude - 15°												
25	1.15	1.11	1.06	1.01	.98	.96	.97	1.00	1.04	1.09	1.14	1.17
30	1.27	1.18	1.10	1.02	.97	.95	.96	1.00	1.07	1.15	1.24	1.29
35	1.41	1.28	1.15	1.04	.97	.94	.96	1.01	1.10	1.23	1.37	1.46
40	1.62	1.40	1.21	1.07	.98	.94	.95	1.02	1.15	1.34	1.56	1.70
45	1.92	1.58	1.30	1.10	.98	.94	.96	1.04	1.21	1.48	1.82	2.05
50	2.40	1.83	1.41	1.14	.99	.94	.96	1.07	1.29	1.67	2.22	2.64
55	3.24	2.20	1.57	1.19	1.01	.94	.97	1.10	1.39	1.95	2.89	3.75
Tilt = Latitude												
25	1.33	1.21	1.09	.98	.91	.87	.89	.95	1.05	1.17	1.30	1.37
30	1.46	1.30	1.13	.99	.90	.86	.88	.95	1.08	1.24	1.41	1.51
35	1.63	1.40	1.19	1.01	.90	.85	.87	.96	1.11	1.33	1.57	1.71
40	1.88	1.55	1.26	1.03	.90	.85	.87	.97	1.16	1.44	1.78	1.99
45	2.23	1.74	1.35	1.06	.91	.85	.87	.99	1.22	1.59	2.08	2.41
50	2.78	2.02	1.47	1.10	.92	.85	.88	1.01	1.30	1.81	2.54	3.10
55	3.76	2.43	1.63	1.15	.93	.85	.88	1.05	1.40	2.11	3.31	4.41
Tilt = Latitude + 15°												
25	1.43	1.26	1.07	.91	.80	.75	.77	.86	1.00	1.19	1.39	1.49
30	1.57	1.34	1.11	.92	.79	.74	.76	.86	1.03	1.26	1.51	1.65
35	1.76	1.45	1.16	.93	.79	.73	.76	.86	1.06	1.35	1.67	1.86
40	2.02	1.60	1.23	.95	.79	.73	.75	.87	1.11	1.47	1.90	2.17
45	2.40	1.80	1.32	.98	.79	.72	.75	.89	1.16	1.62	2.22	2.62
50	2.99	2.09	1.44	1.01	.80	.72	.75	.91	1.24	1.84	2.70	3.37
55	4.04	2.52	1.59	1.05	.81	.72	.76	.93	1.34	2.15	3.52	4.78
Tilt = Vertical												
25	1.20	.92	.63	.39	.28	.25	.26	.34	.53	.82	1.12	1.28
30	1.37	1.04	.72	.46	.32	.28	.30	.39	.60	.93	1.28	1.48
35	1.59	1.19	.81	.52	.37	.32	.34	.45	.68	1.06	1.48	1.73
40	1.88	1.37	.92	.59	.42	.37	.39	.51	.77	1.21	1.73	2.07
45	2.30	1.61	1.05	.66	.48	.42	.44	.58	.87	1.40	2.09	2.56
50	2.93	1.93	1.21	.75	.54	.47	.50	.65	.99	1.65	2.61	3.34
55	4.01	2.39	1.41	.84	.60	.52	.55	.72	1.13	2.00	3.46	4.80

Table A3-7. Ratio of Monthly Average Daily Radiation on a Tilted Surface to that on a Horizontal Surface [Ref. 7]

 \bar{R} for $\bar{K}_t = .70$

Latitude	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Tilt = Latitude -15°												
25	1.17	1.12	1.06	1.01	.98	.96	.97	1.00	1.04	1.10	1.16	1.19
30	1.30	1.20	1.11	1.03	.97	.95	.96	1.00	1.07	1.17	1.27	1.33
35	1.46	1.31	1.17	1.05	.97	.94	.95	1.01	1.12	1.26	1.42	1.51
40	1.69	1.45	1.24	1.08	.98	.94	.95	1.03	1.17	1.38	1.62	1.78
45	2.03	1.65	1.34	1.11	.99	.94	.96	1.06	1.24	1.53	1.92	2.18
50	2.56	1.93	1.47	1.16	1.00	.94	.97	1.09	1.33	1.75	2.36	2.83
55	3.50	2.34	1.64	1.22	1.02	.94	.98	1.13	1.45	2.06	3.11	4.06
Tilt = Latitude												
25	1.37	1.24	1.11	.98	.90	.86	.88	.95	1.06	1.20	1.34	1.41
30	1.52	1.34	1.16	1.00	.90	.85	.87	.95	1.09	1.27	1.47	1.58
35	1.71	1.46	1.22	1.02	.90	.85	.87	.96	1.13	1.37	1.64	1.80
40	1.98	1.62	1.30	1.05	.90	.84	.87	.98	1.19	1.50	1.88	2.11
45	2.38	1.84	1.40	.98	.91	.84	.87	1.00	1.26	1.67	2.21	2.58
50	3.00	2.15	1.53	1.13	.92	.85	.88	1.03	1.35	1.91	2.73	3.35
55	4.09	2.61	1.72	1.19	.94	.85	.89	1.07	1.47	2.25	3.58	4.80
Tilt = Latitude + 15°												
25	1.49	1.30	1.09	.91	.78	.73	.76	.85	1.01	1.23	1.44	1.56
30	1.65	1.39	1.14	.92	.78	.72	.75	.86	1.04	1.30	1.58	1.73
35	1.86	1.52	1.20	.94	.78	.72	.75	.87	1.09	1.41	1.76	1.98
40	2.15	1.69	1.28	.96	.78	.72	.74	.88	1.14	1.54	2.01	2.32
45	2.58	1.91	1.38	.99	.79	.71	.75	.90	1.20	1.71	2.37	2.83
50	3.24	2.23	1.51	1.04	.80	.72	.75	.92	1.29	1.96	2.92	3.66
55	4.41	2.71	1.69	1.09	.81	.72	.76	.96	1.40	2.31	3.83	5.23
Tilt = Vertical												
25	1.26	.96	.64	.37	.25	.21	.23	.31	.52	.85	1.18	1.36
30	1.46	1.09	.73	.44	.29	.25	.27	.37	.60	.97	1.35	1.57
35	1.70	1.26	.84	.51	.35	.29	.32	.43	.69	1.11	1.57	1.85
40	2.03	1.46	.96	.59	.41	.34	.37	.50	.79	1.28	1.86	2.23
45	2.48	1.72	1.10	.67	.47	.40	.43	.57	.90	1.49	2.25	2.77
50	3.18	2.07	1.27	.76	.53	.45	.48	.65	1.03	1.77	2.83	3.65
55	4.39	2.59	1.50	.87	.60	.51	.55	.73	1.19	2.15	3.78	5.26

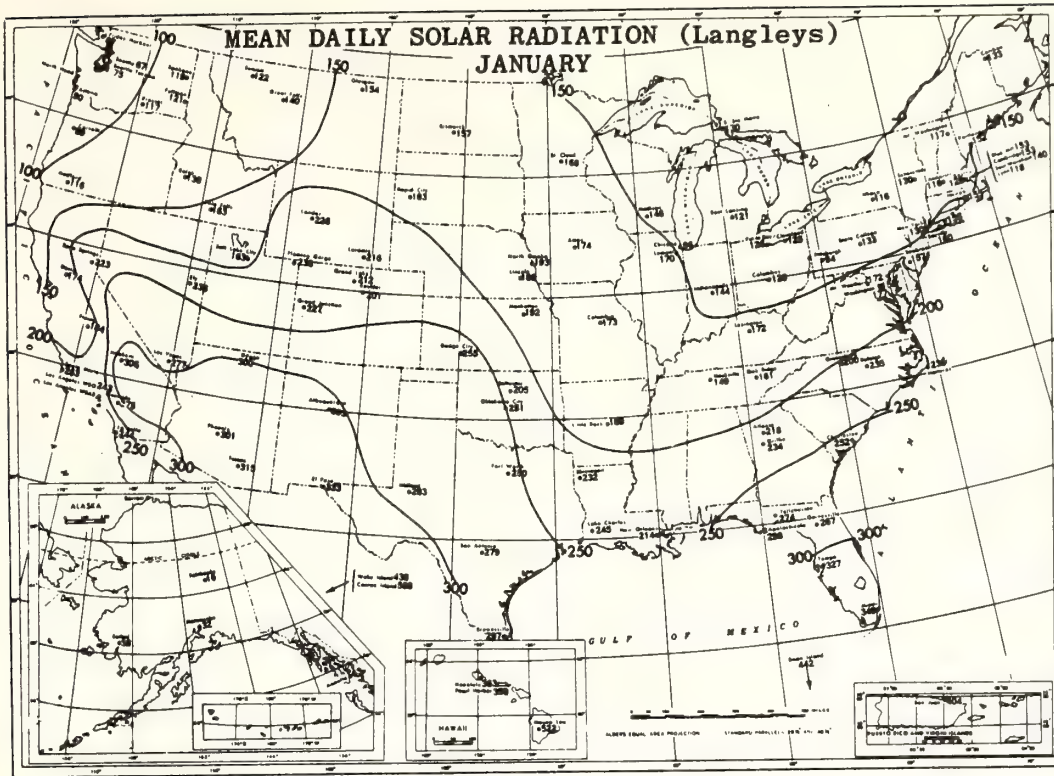


Figure A3-1. Mean Daily Solar Radiation (Langley), January

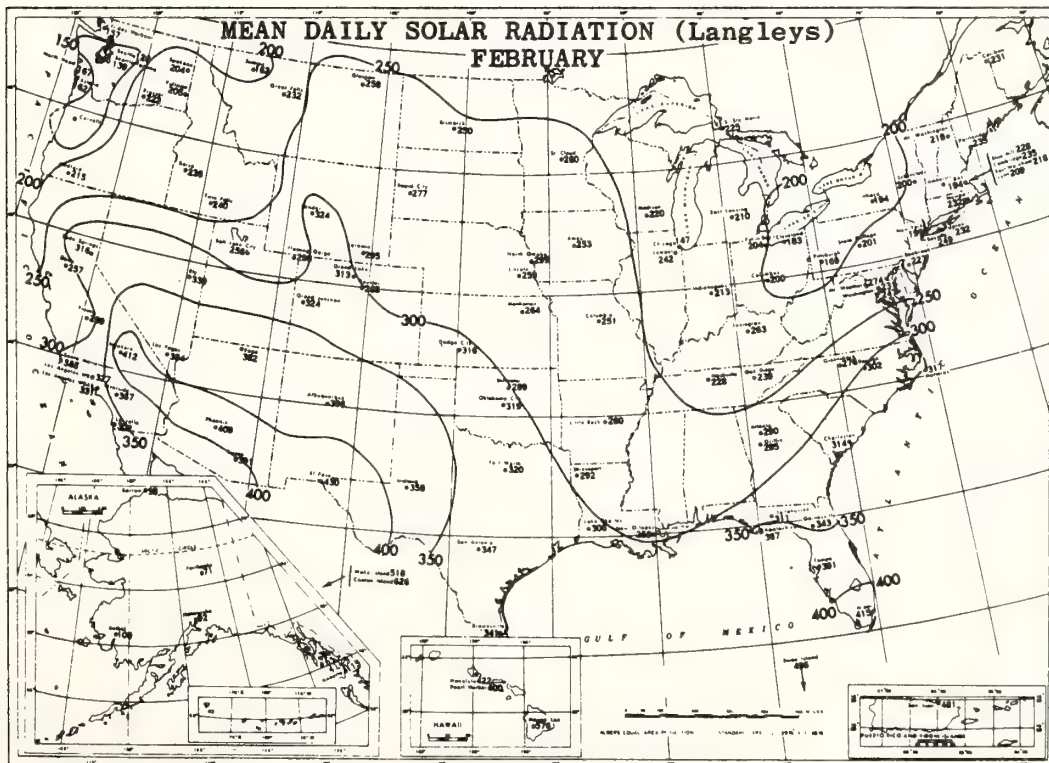


Figure A3-2. Mean Daily Solar Radiation (Langley), February

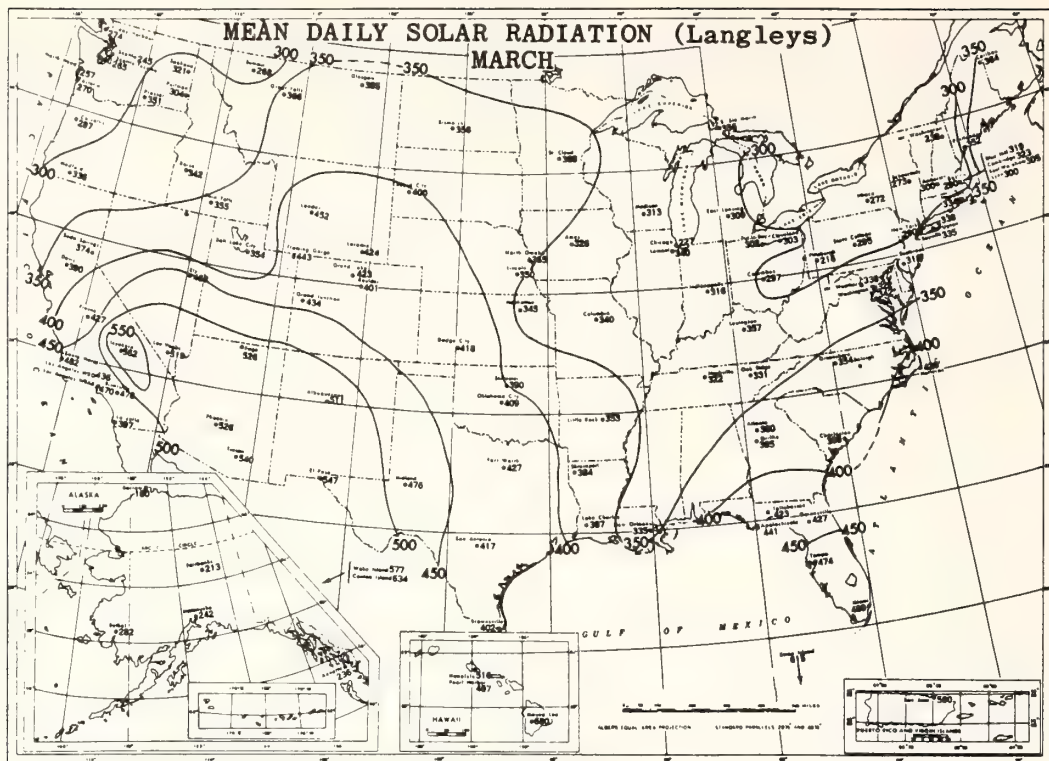


Figure A3-3. Mean Daily Solar Radiation (Langleys), March

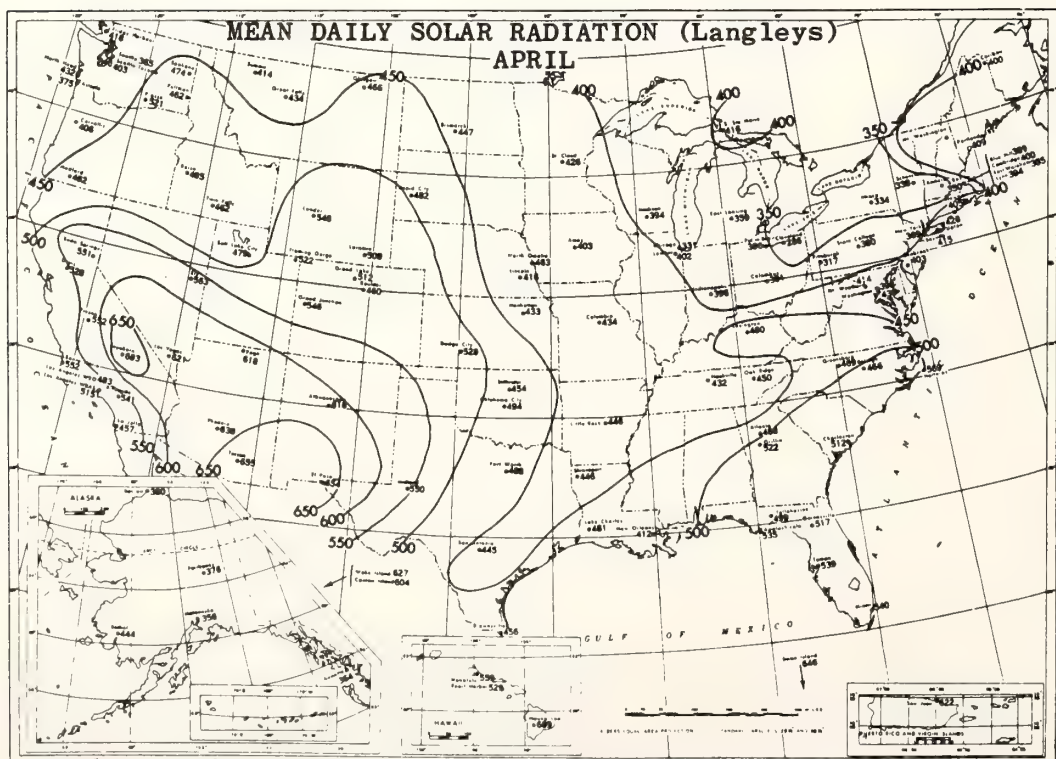


Figure A3-4. Mean Daily Solar Radiation (Langleys), April

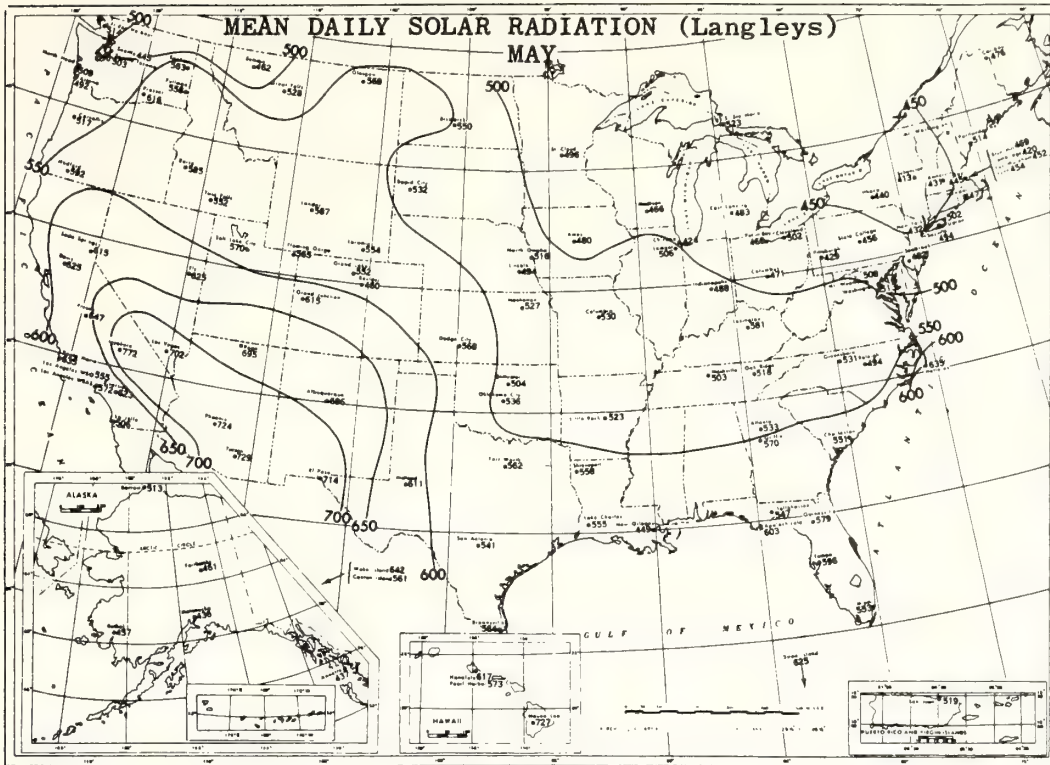


Figure A3-5. Mean Daily Solar Radiation (Langleys), May

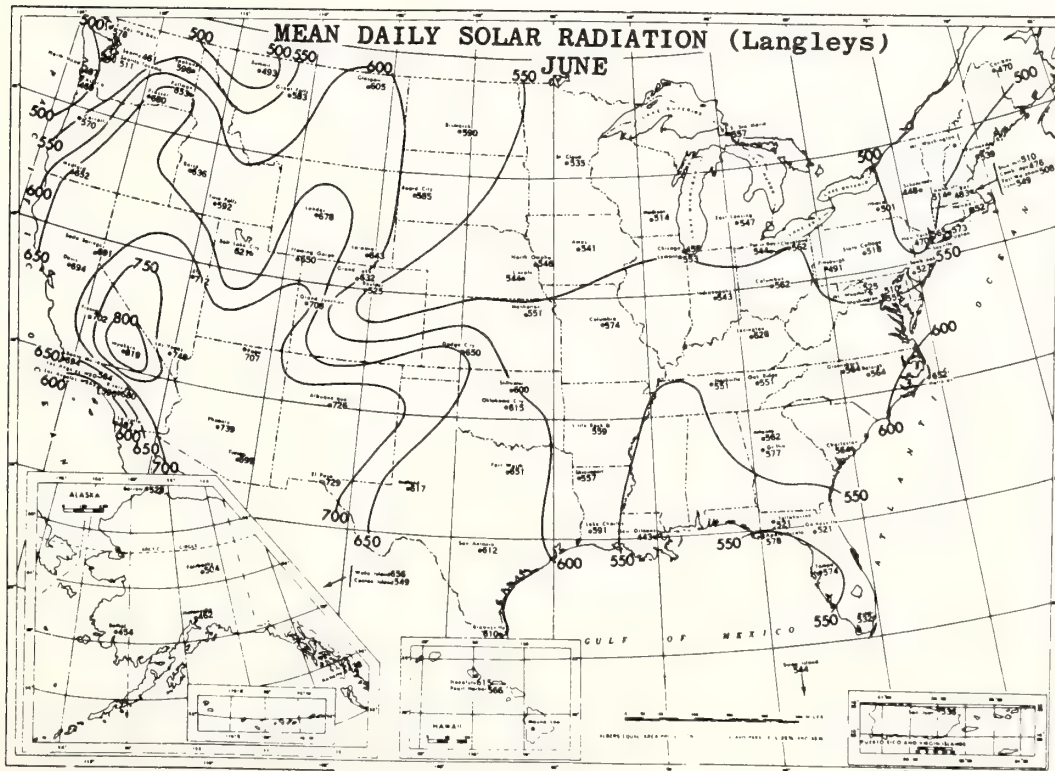


Figure A3-6. Mean Daily Solar Radiation (Langleys), June

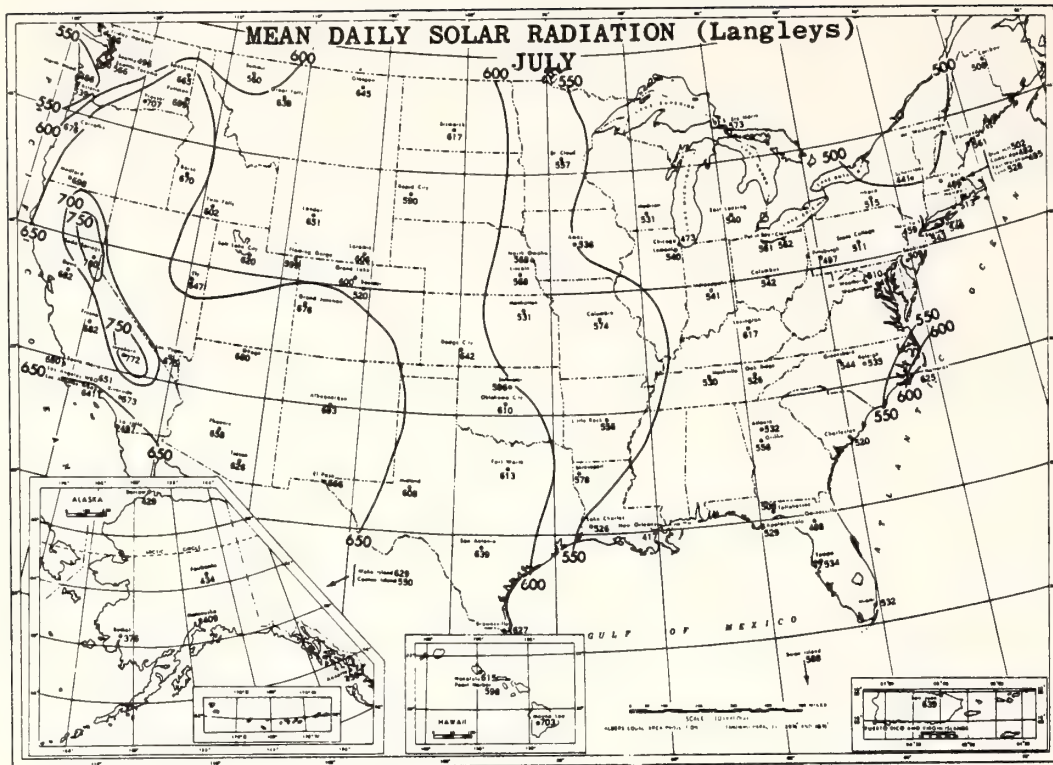


Figure A3-7. Mean Daily Solar Radiation (Langley), July

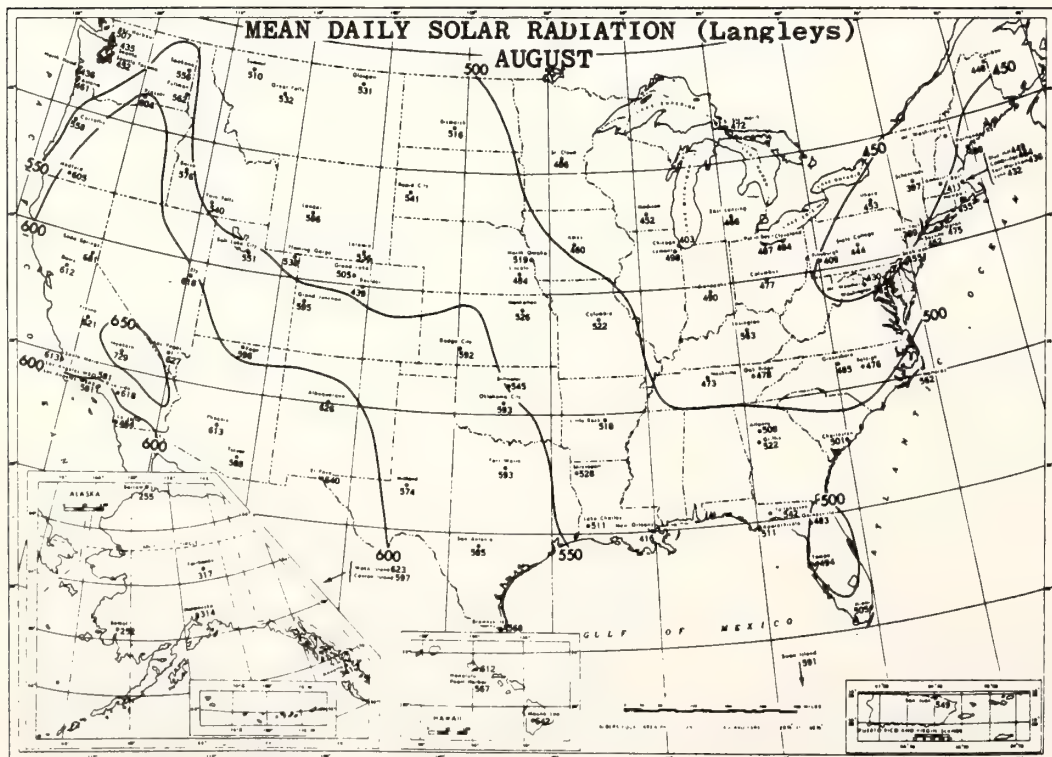


Figure A3-8. Mean Daily Solar Radiation (Langley), August

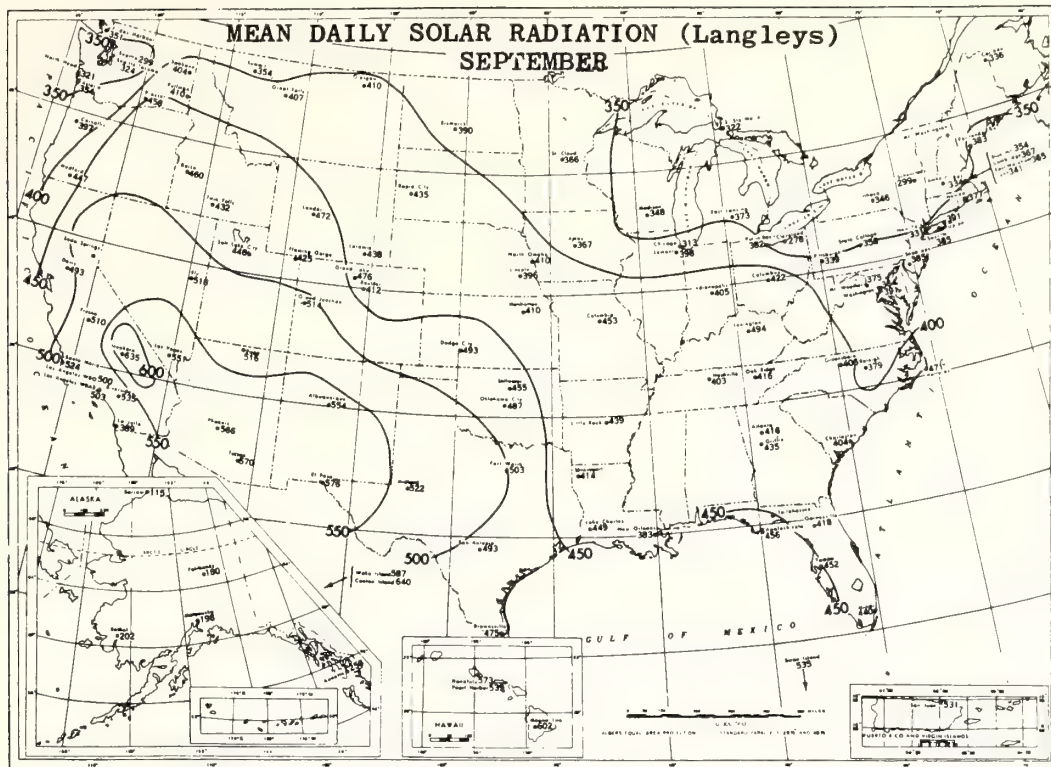


Figure A3-9. Mean Daily Solar Radiation (Langley), September

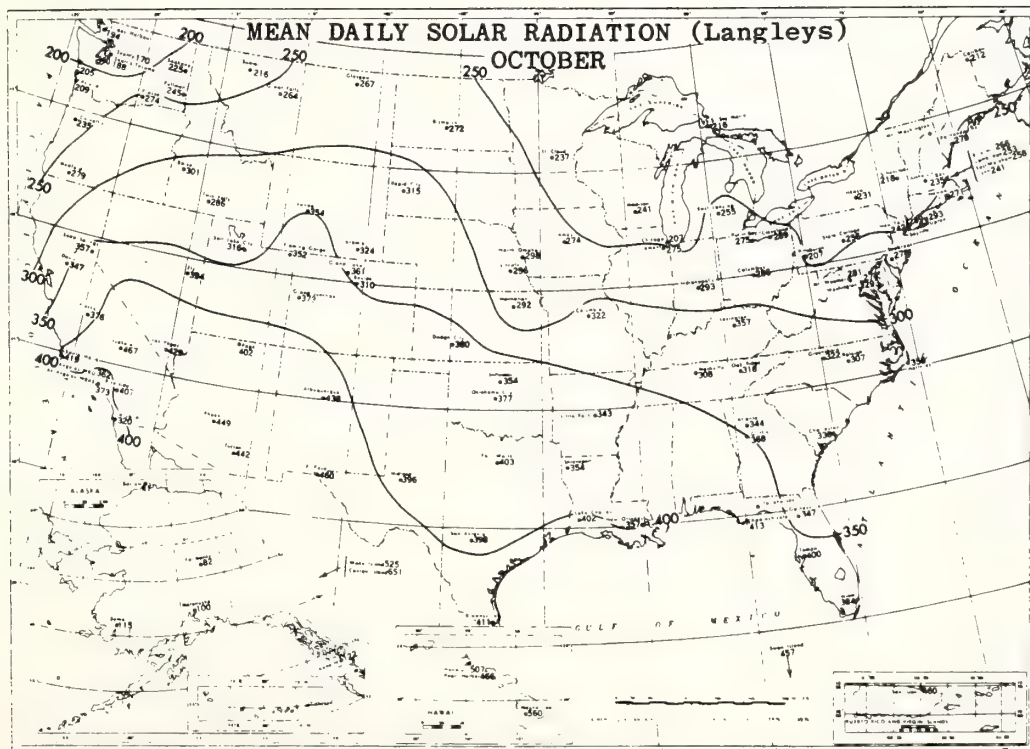


Figure A3-10. Mean Daily Solar Radiation (Langley), October

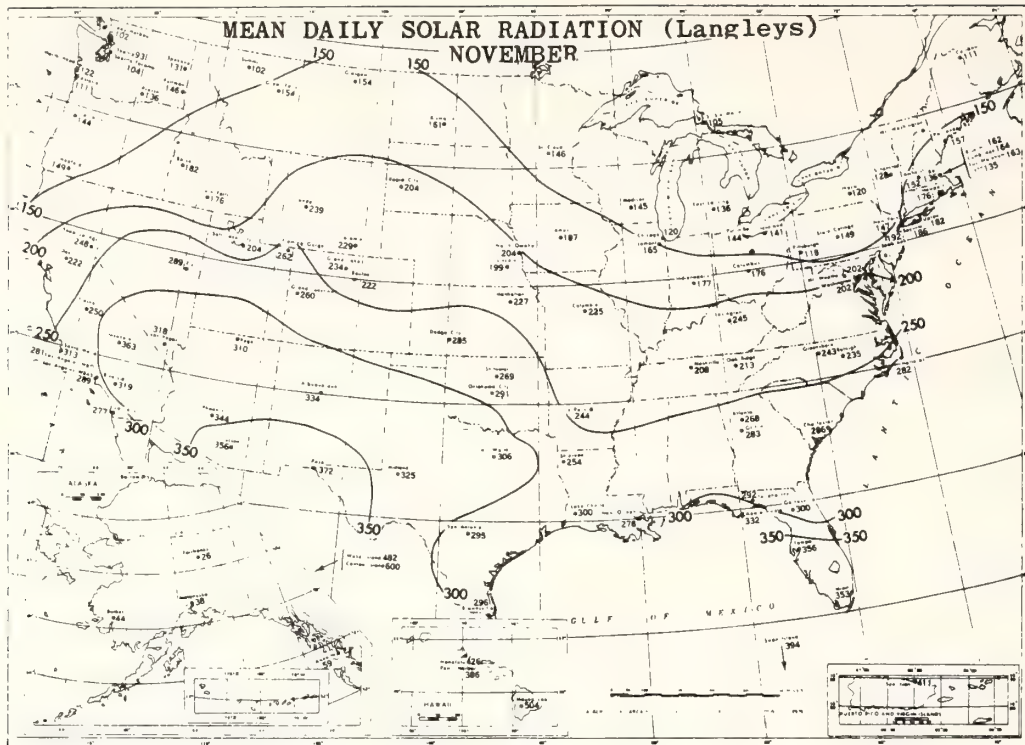


Figure A3-11. Mean Daily Solar Radiation (Langleys), November

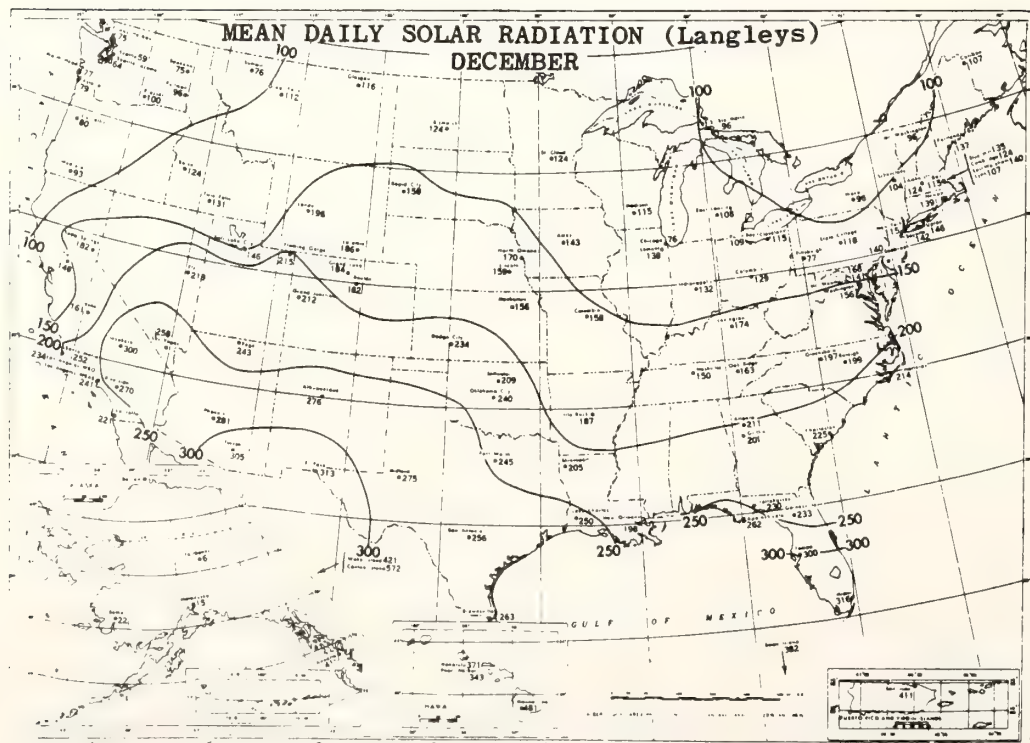


Figure A3-12. Mean Daily Solar Radiation (Langleys), December

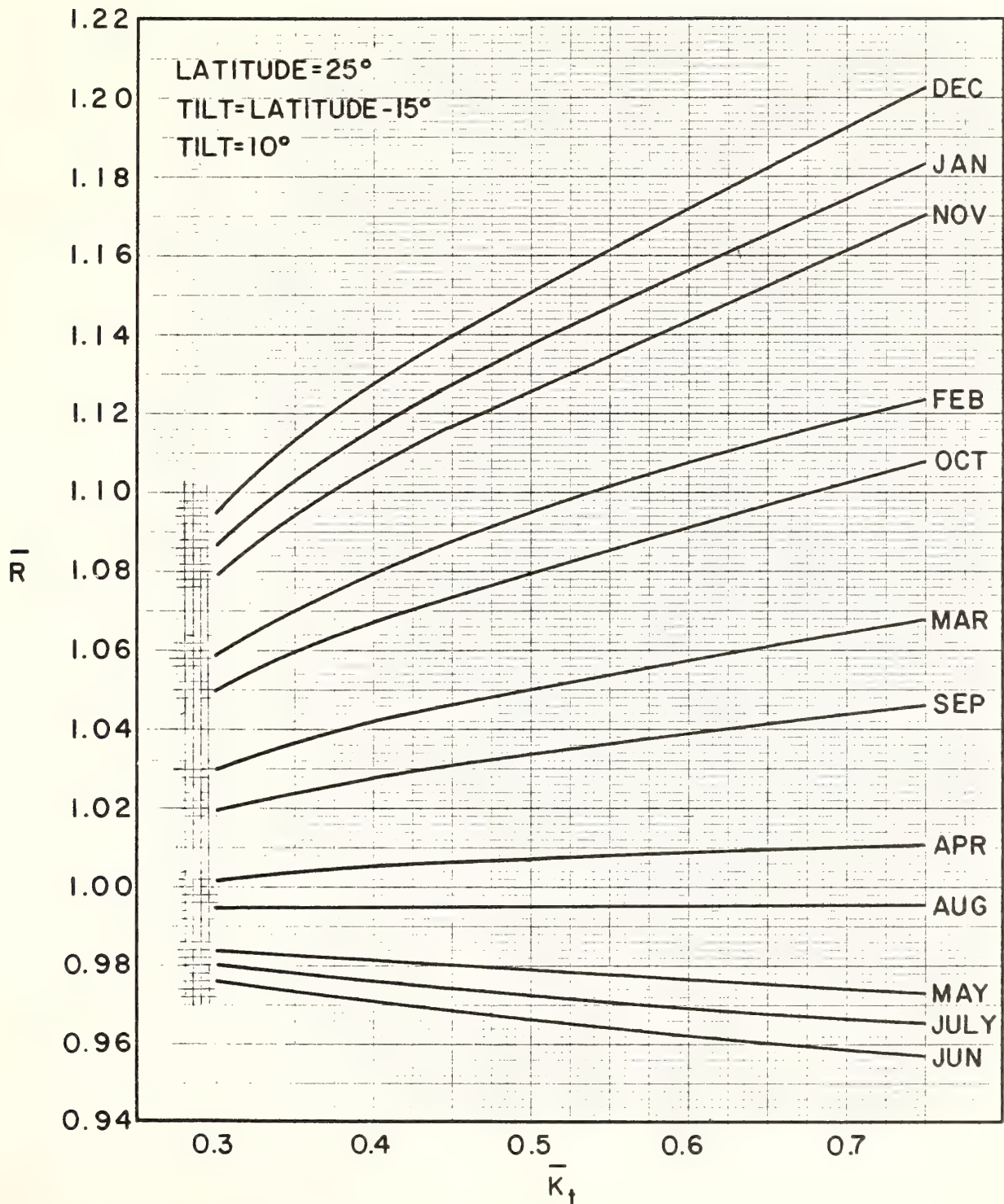
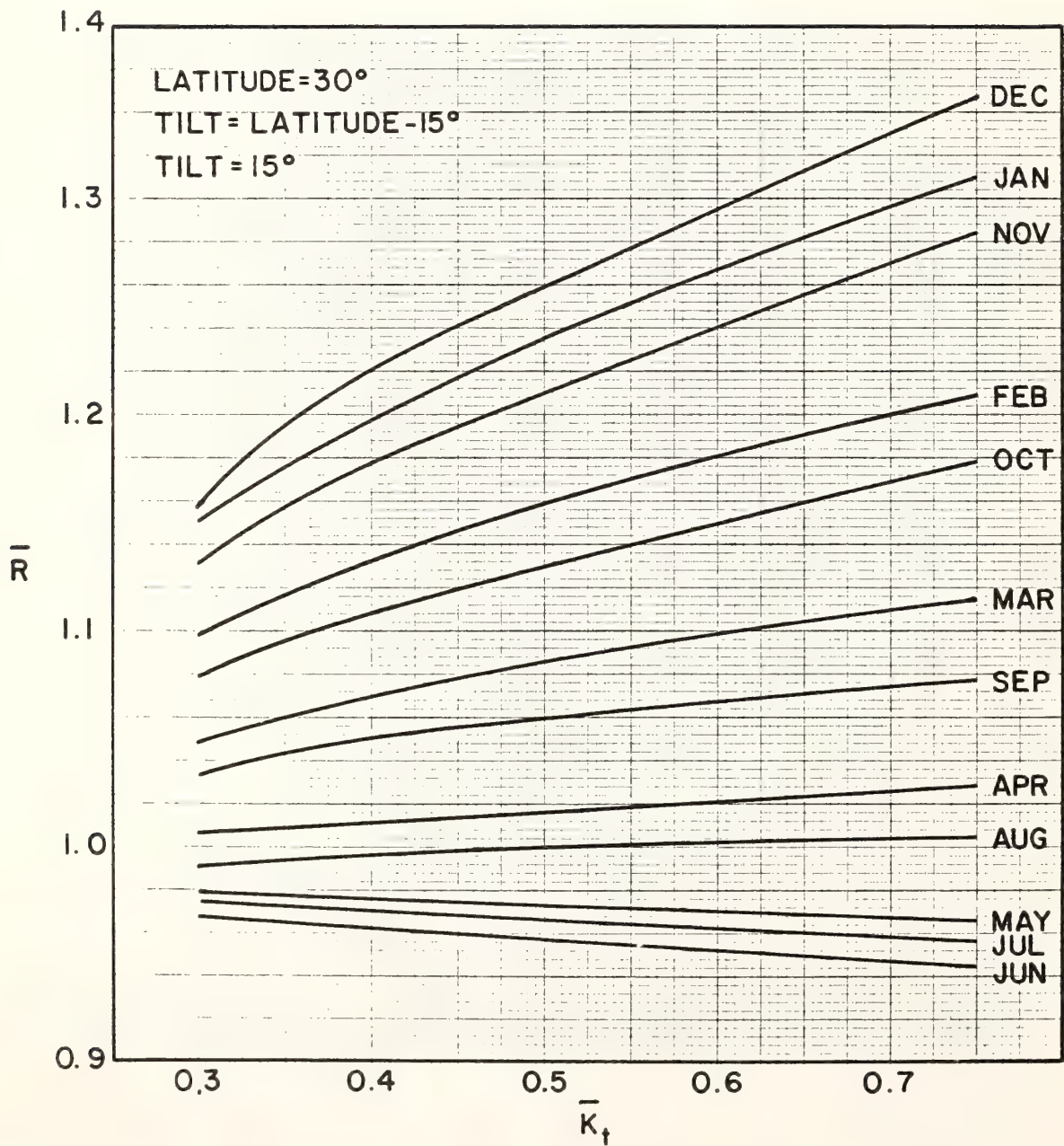
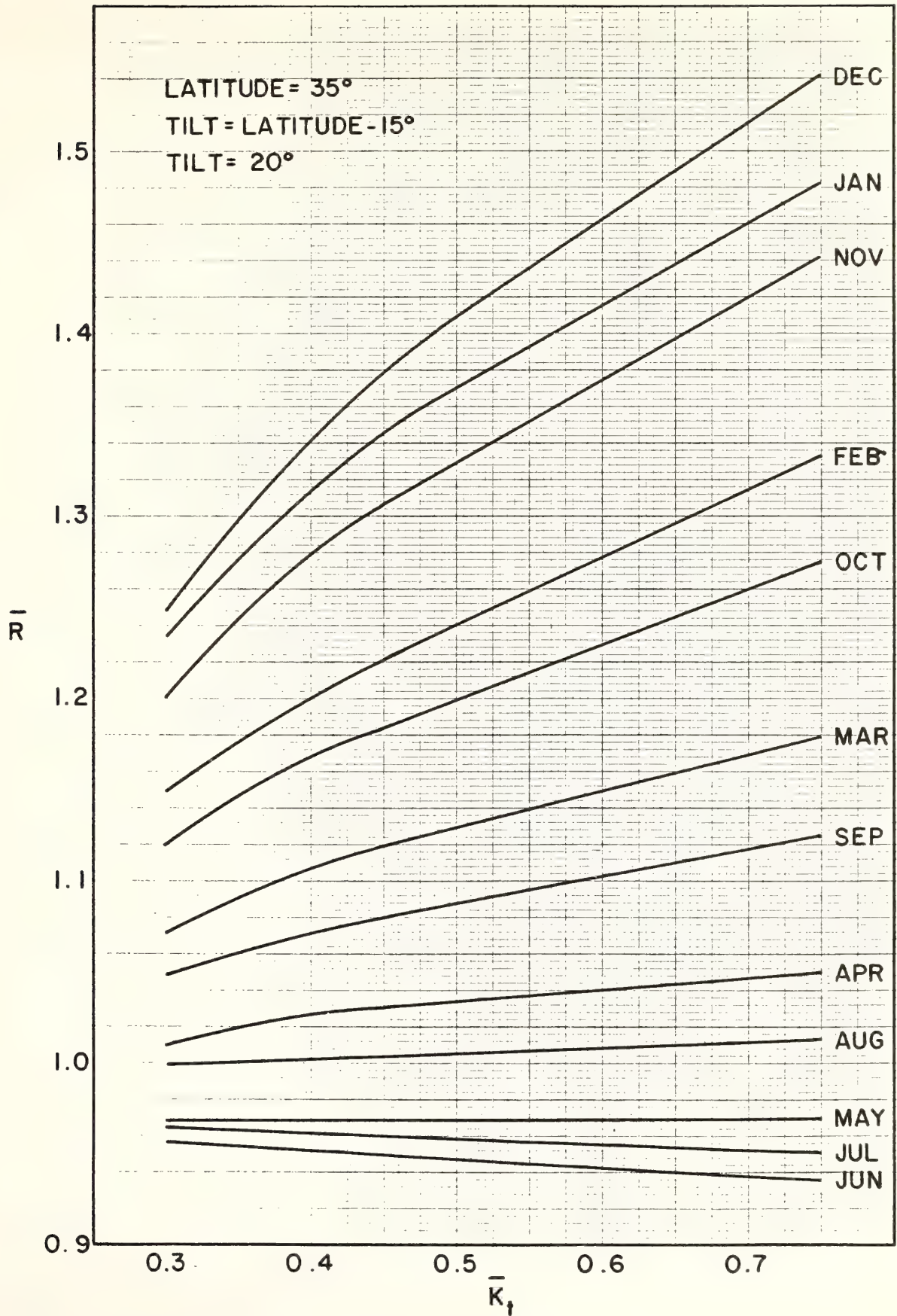


Figure A3-13. Tilt Correction Factor for 25° Latitude, 10° Tilt

Figure A3-14. Tilt Correction Factor for 30° Latitude, 15° Tilt

Figure A3-15. Tilt Correction Factor for 35° Latitude, 20° Tilt

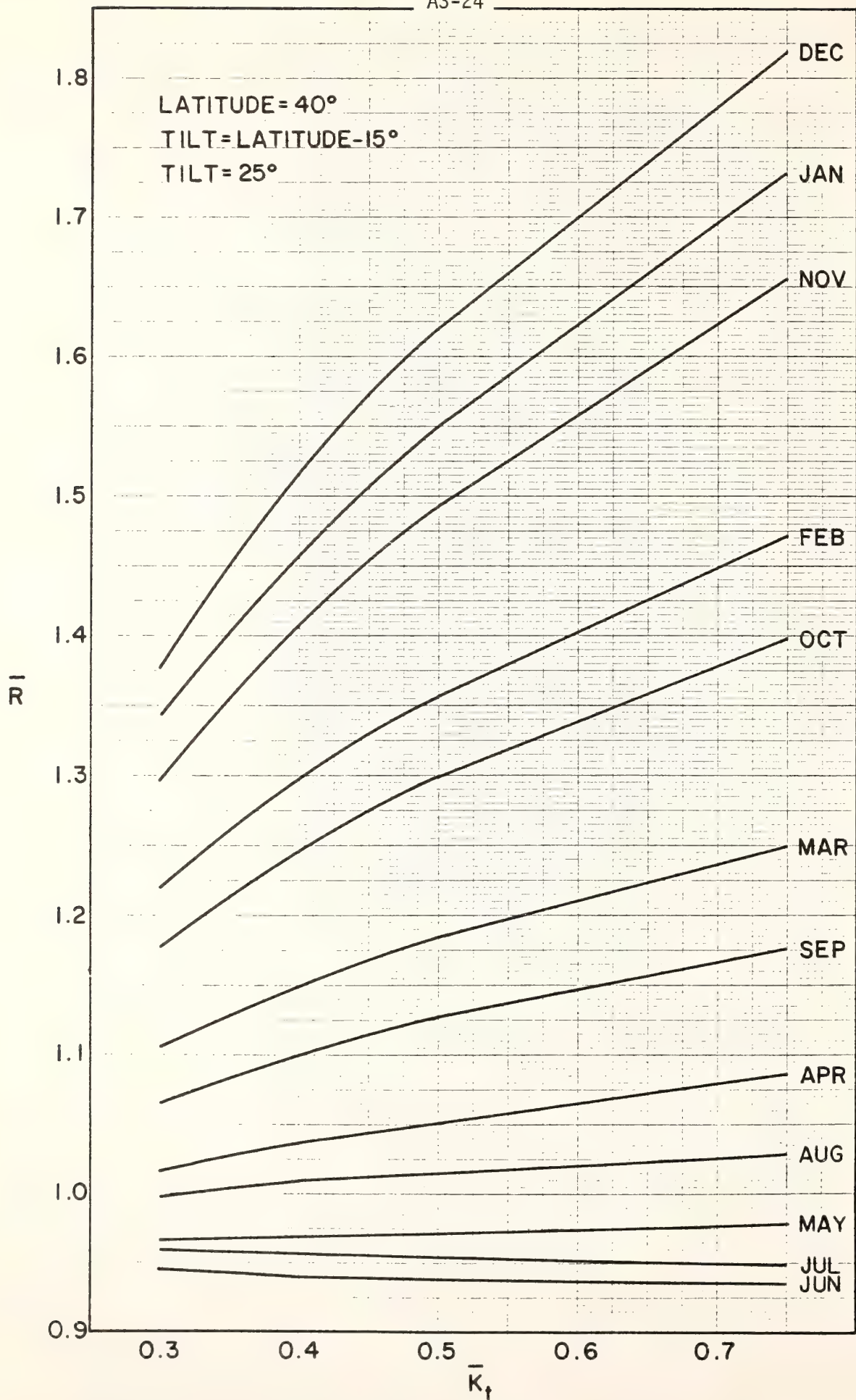


Figure A3-16. Tilt Correction Factor for 40° Latitude, 25° Tilt

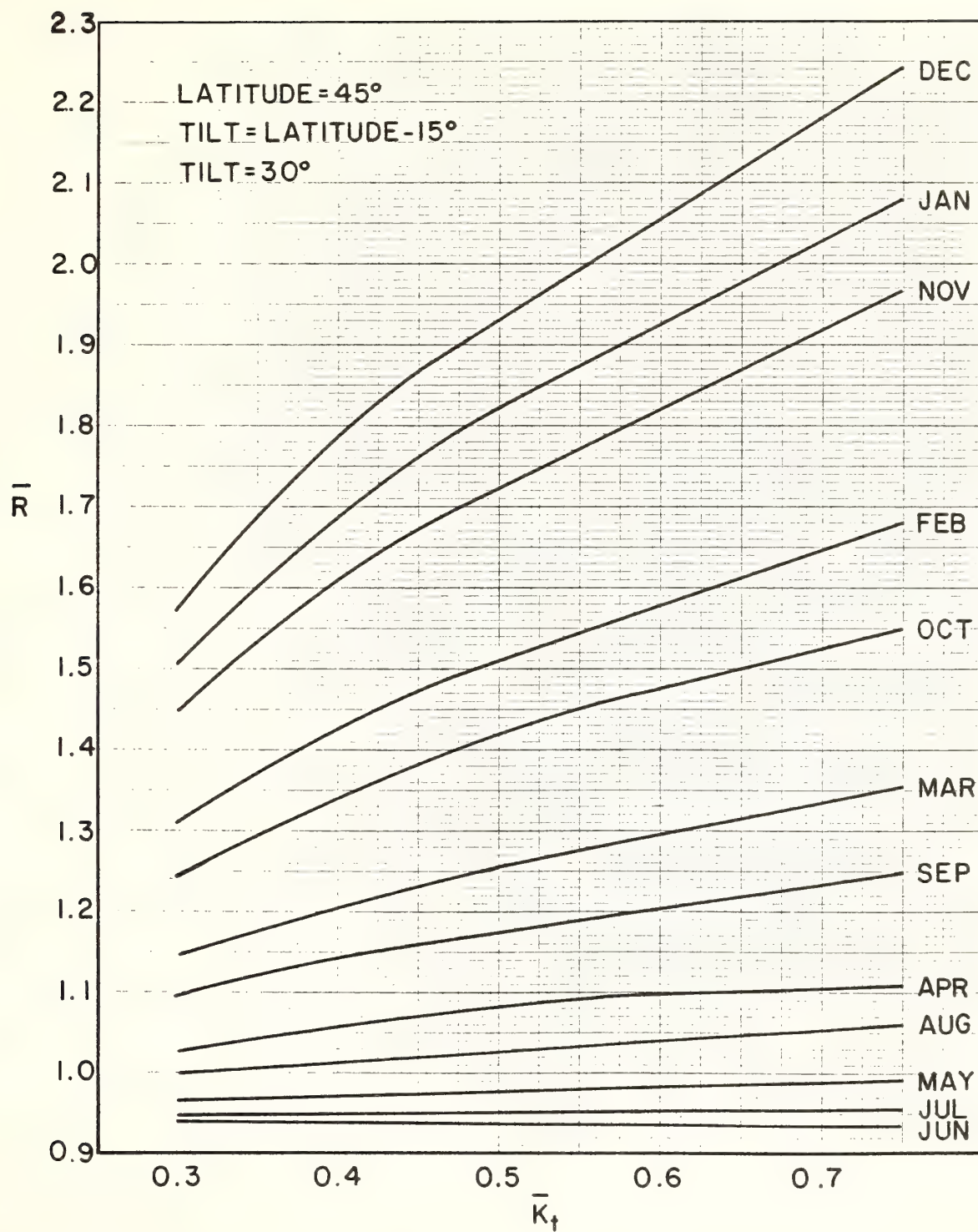


Figure A3-17. Tilt Correction Factor for 45° Latitude, 30° Tilt

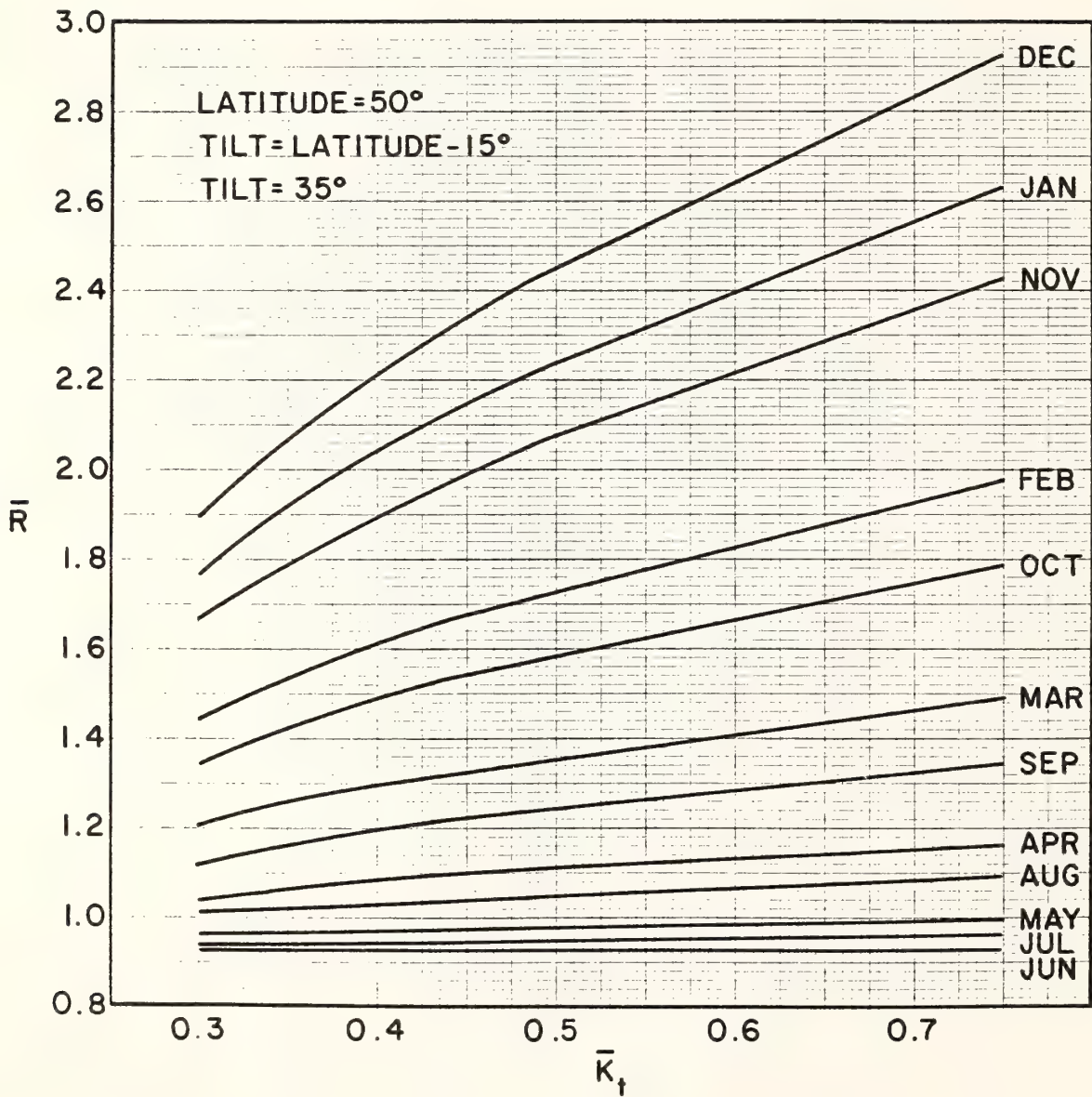


Figure A3-18. Tilt Correction Factor for 50° Latitude, 35° Tilt

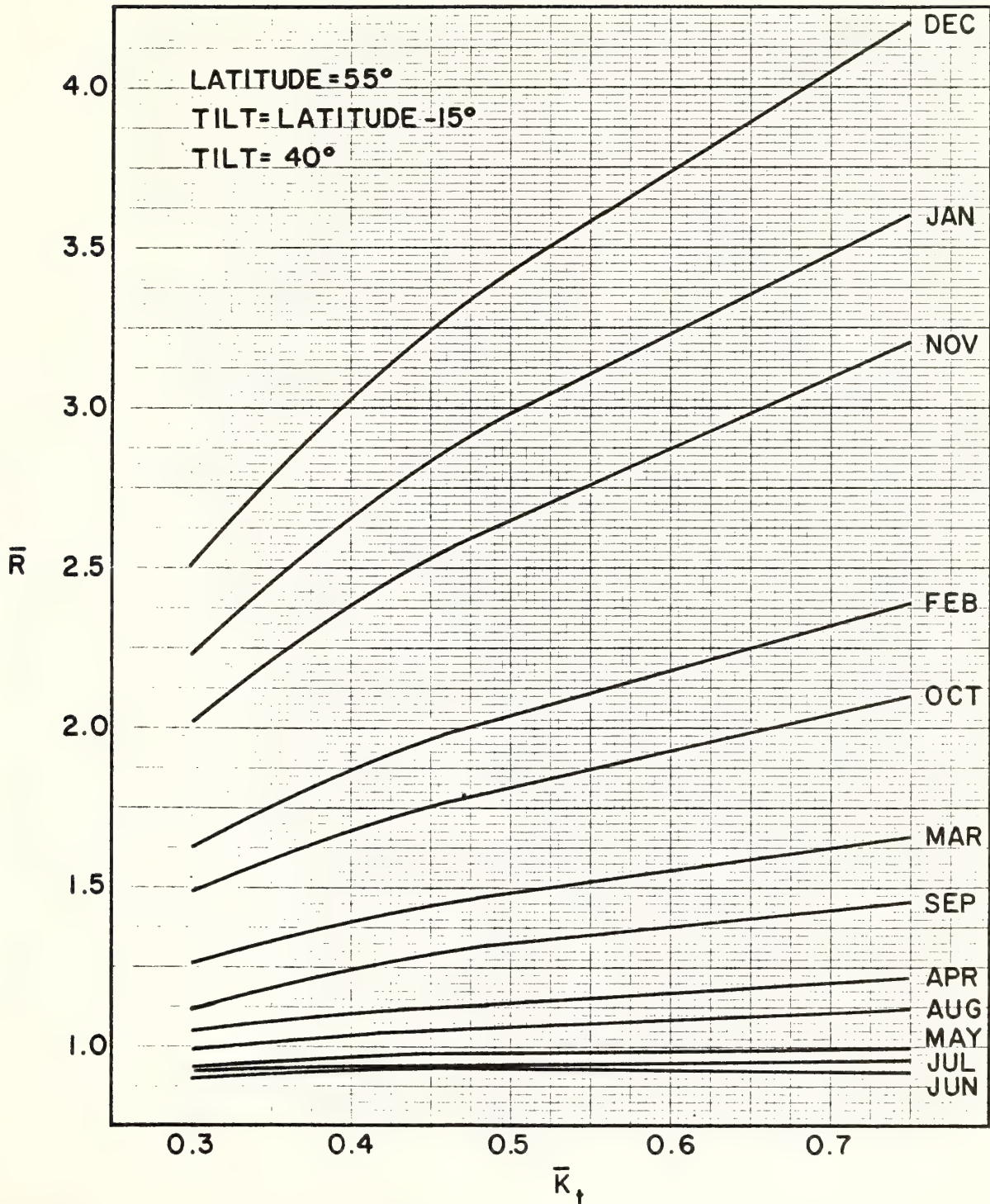


Figure A3-19. Tilt Correction Factor for 55° Latitude, 40° Tilt

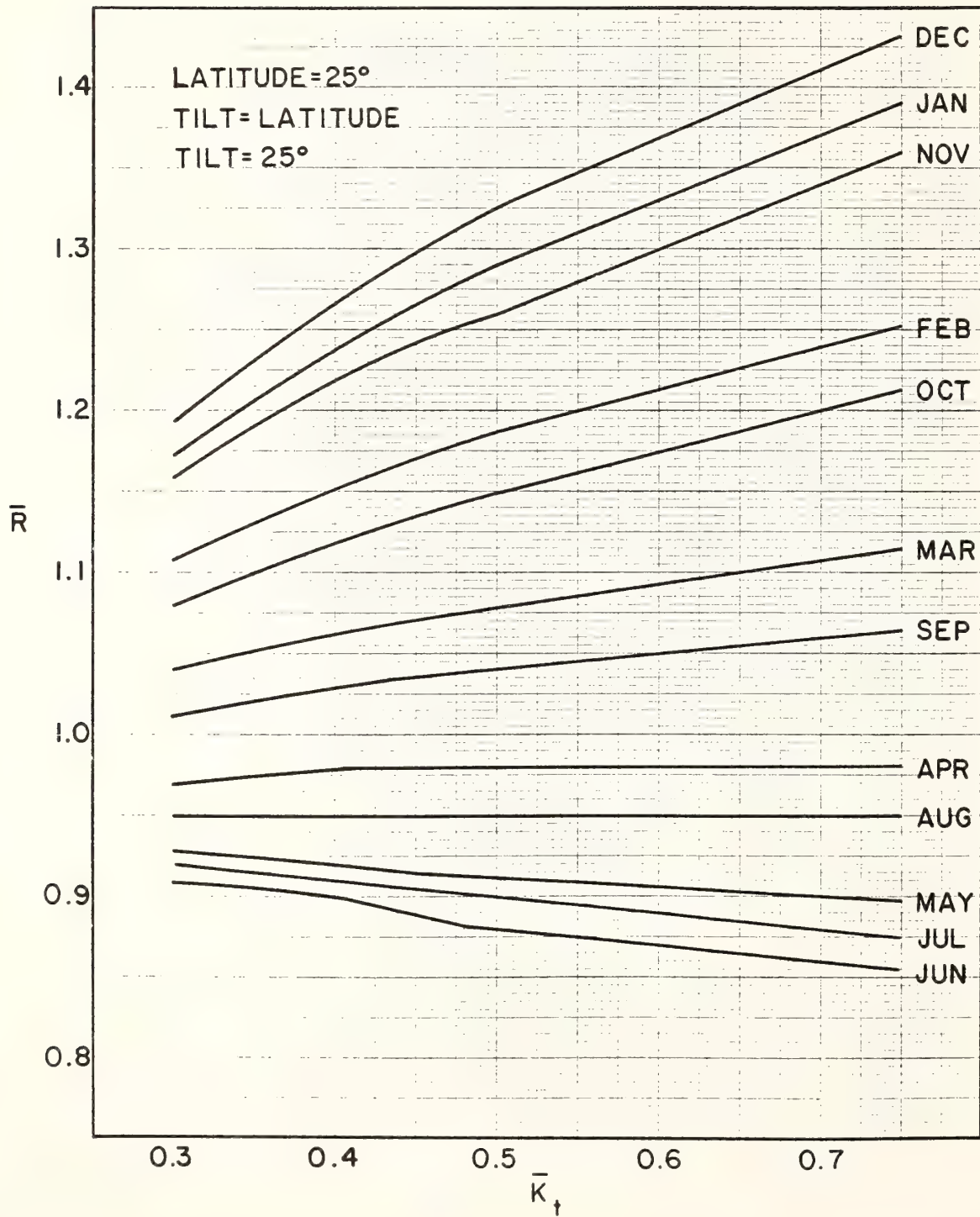


Figure A3-20. Tilt Correction Factor for 25° Latitude, 25° Tilt

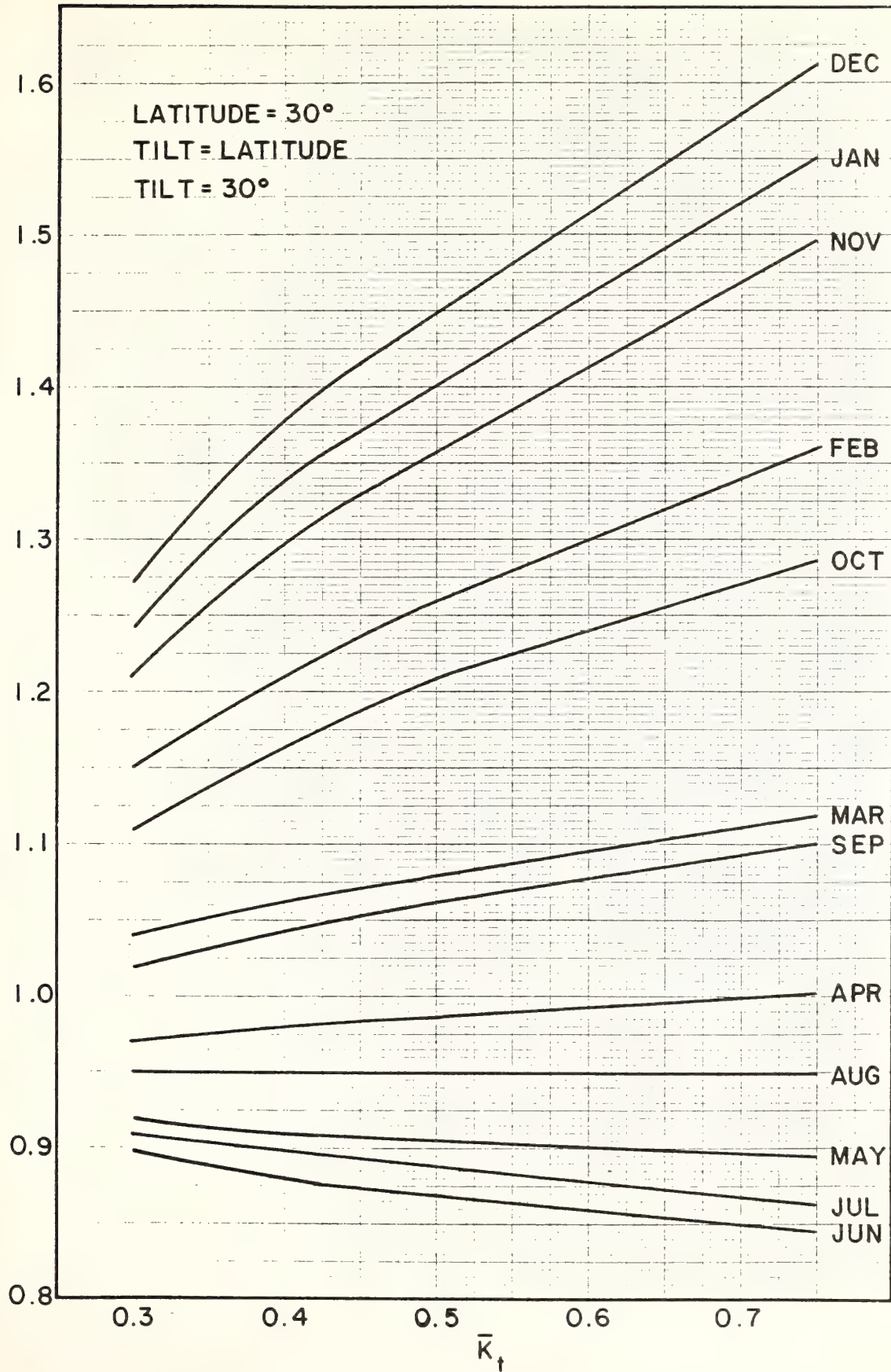


Figure A3-21. Tilt Correction Factor for 30° Latitude, 30° Tilt

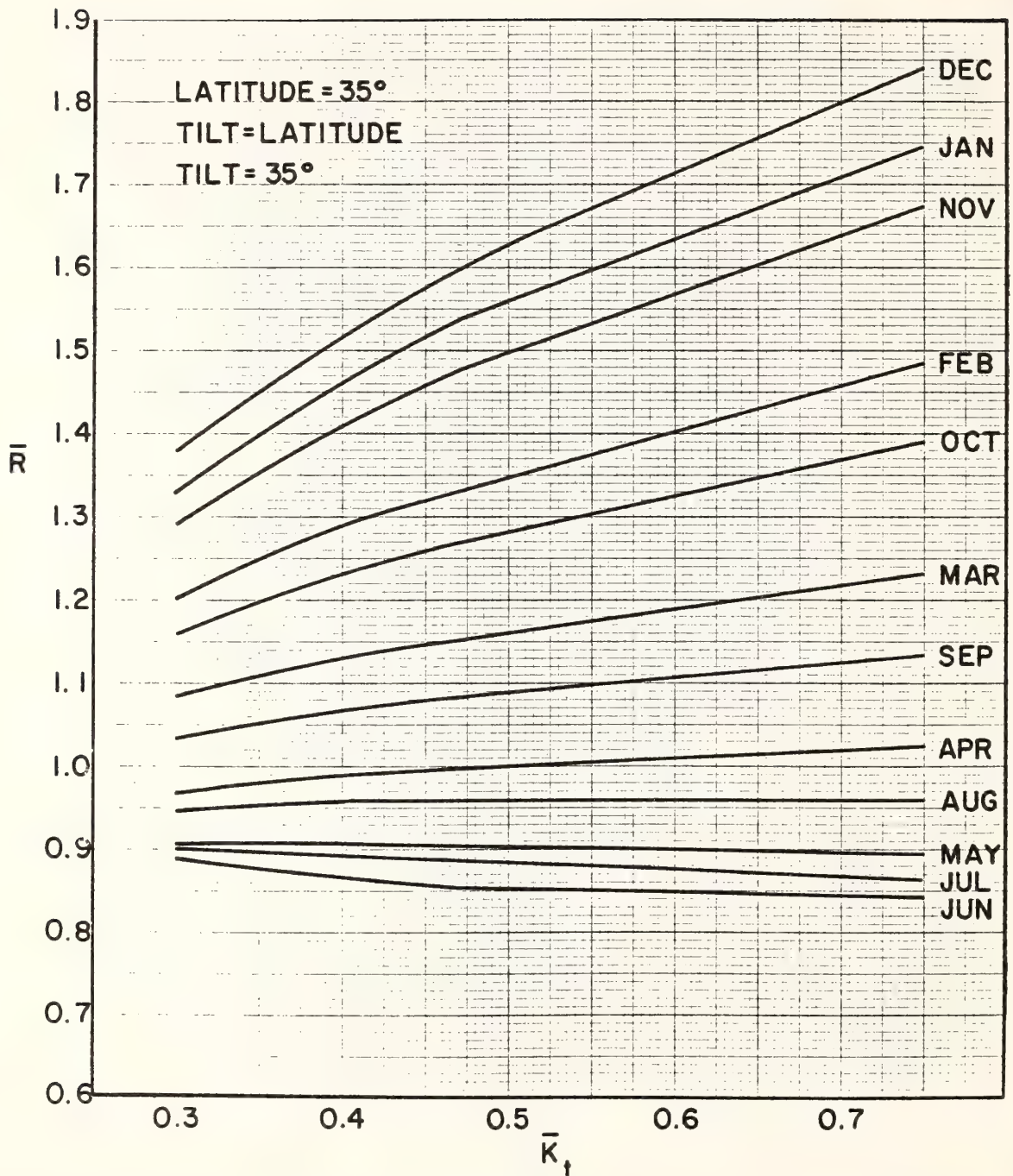


Figure A3-22. Tilt Correction Factor for 35° Latitude, 35° Tilt

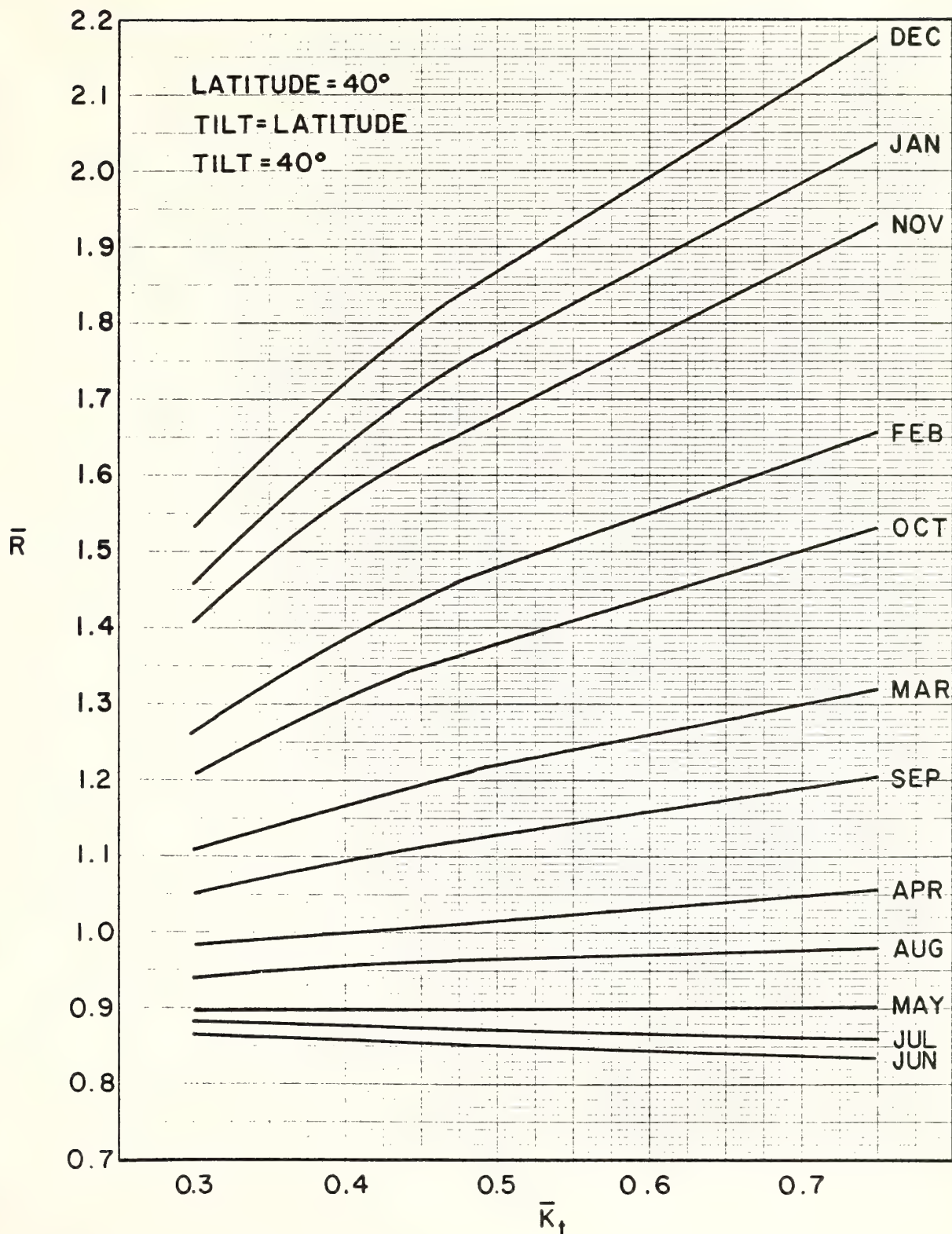


Figure A3-23. Tilt Correction Factor for 40° Latitude, 40° Tilt

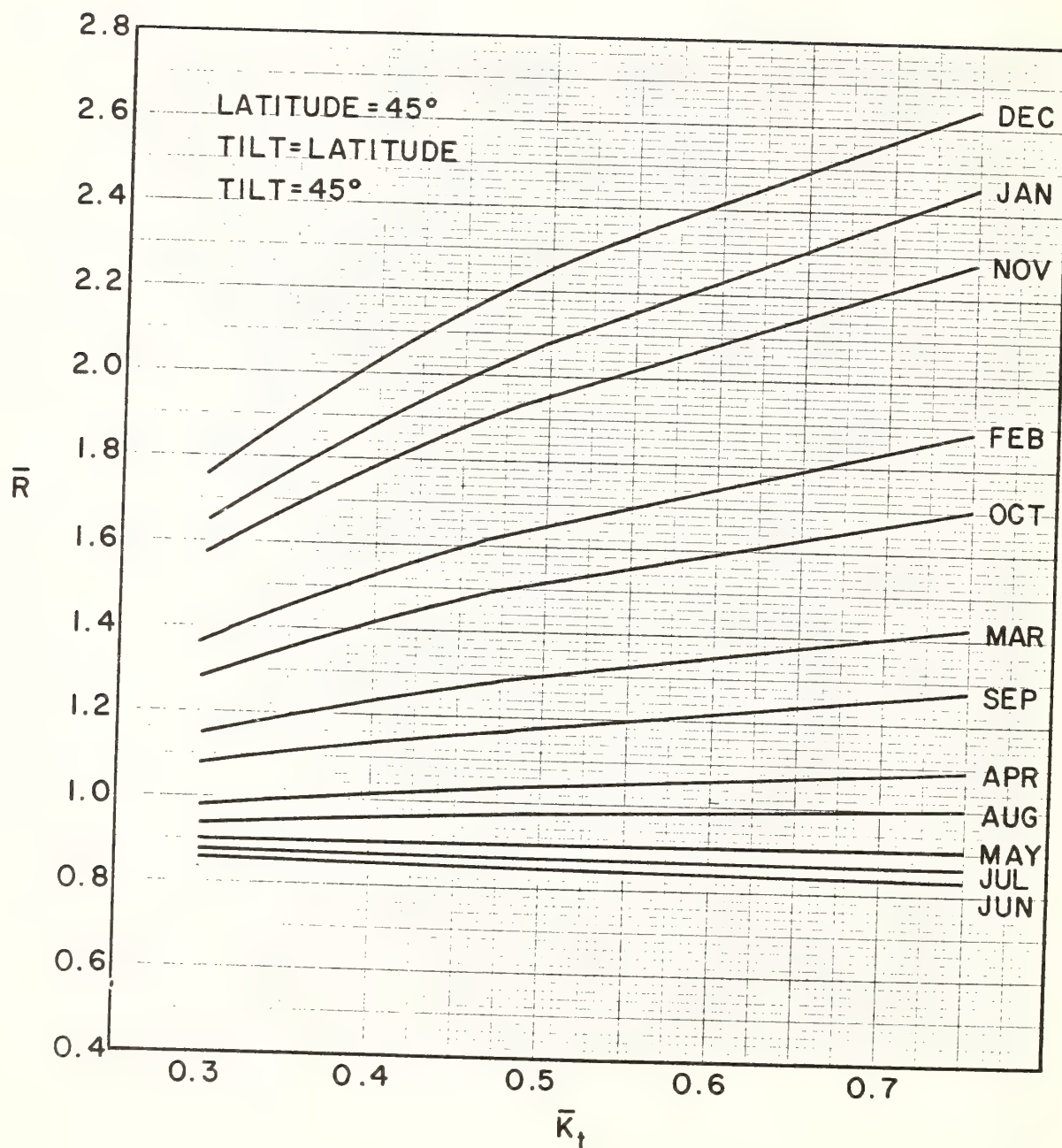


Figure A3-24. Tilt Correction Factor for 45° Latitude, 45° Tilt

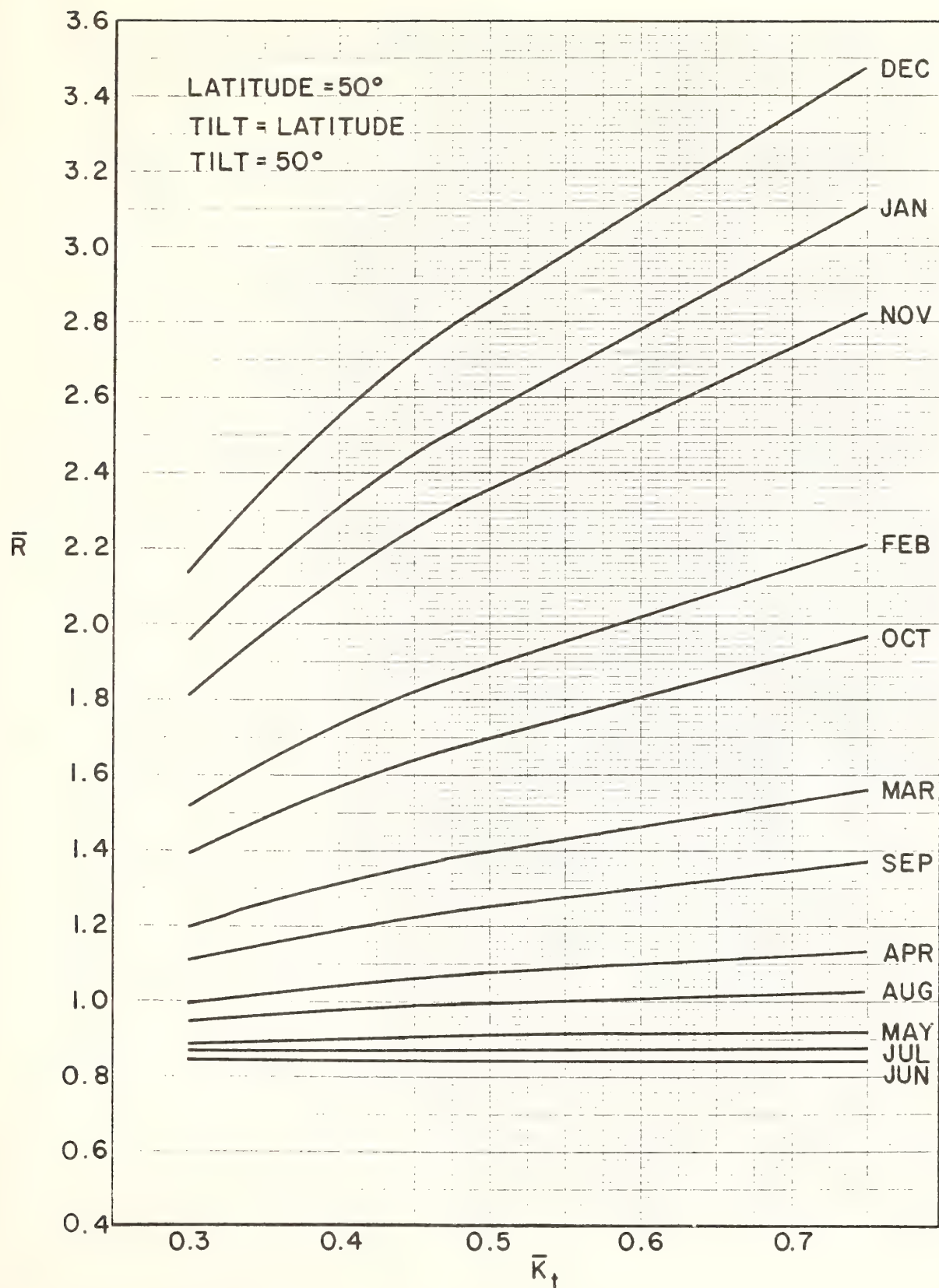


Figure A3-25. Tilt Correction Factor for 50° Latitude, 50° Tilt

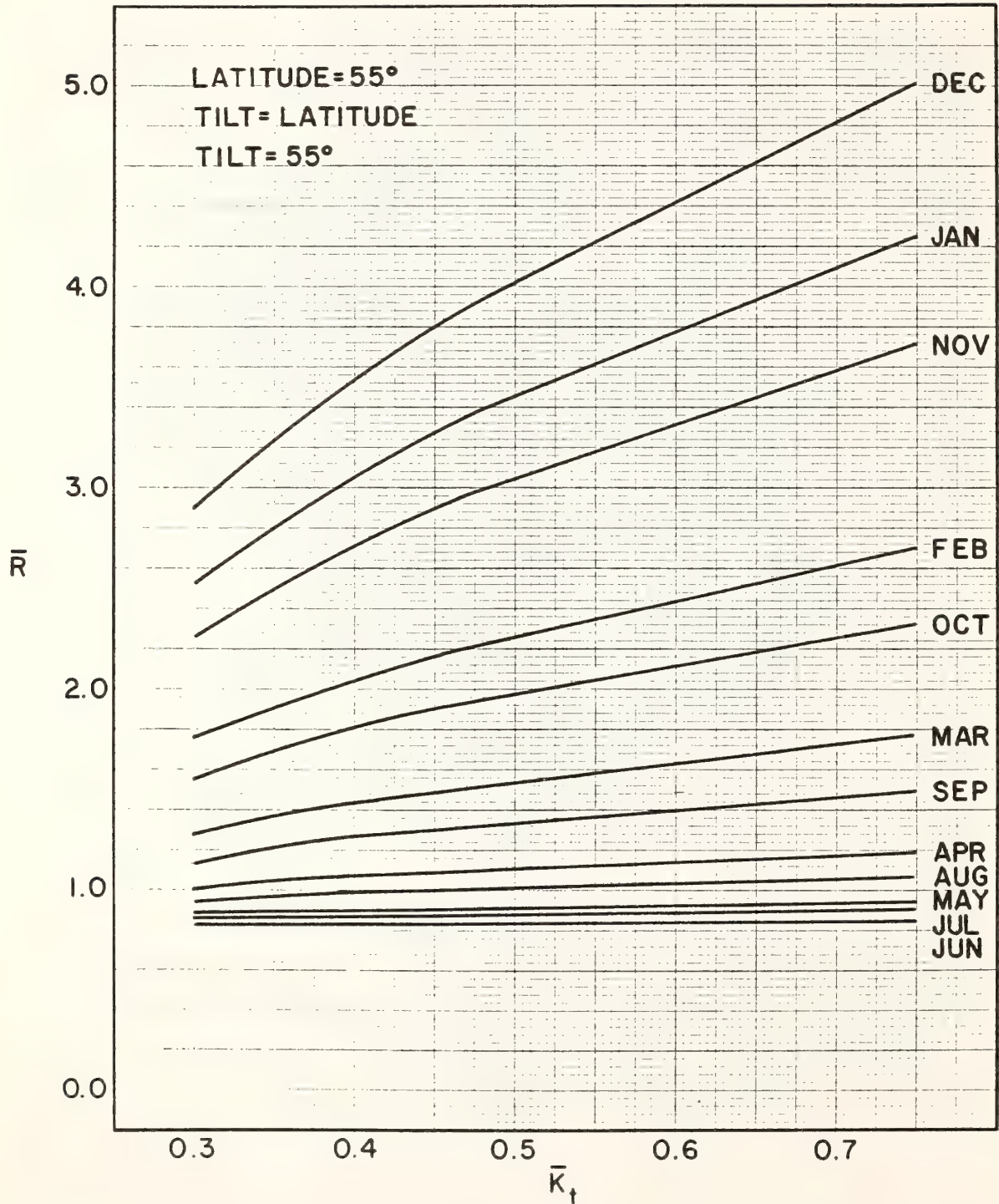


Figure A3-26. Tilt Correction Factor for 55° Latitude, 55° Tilt

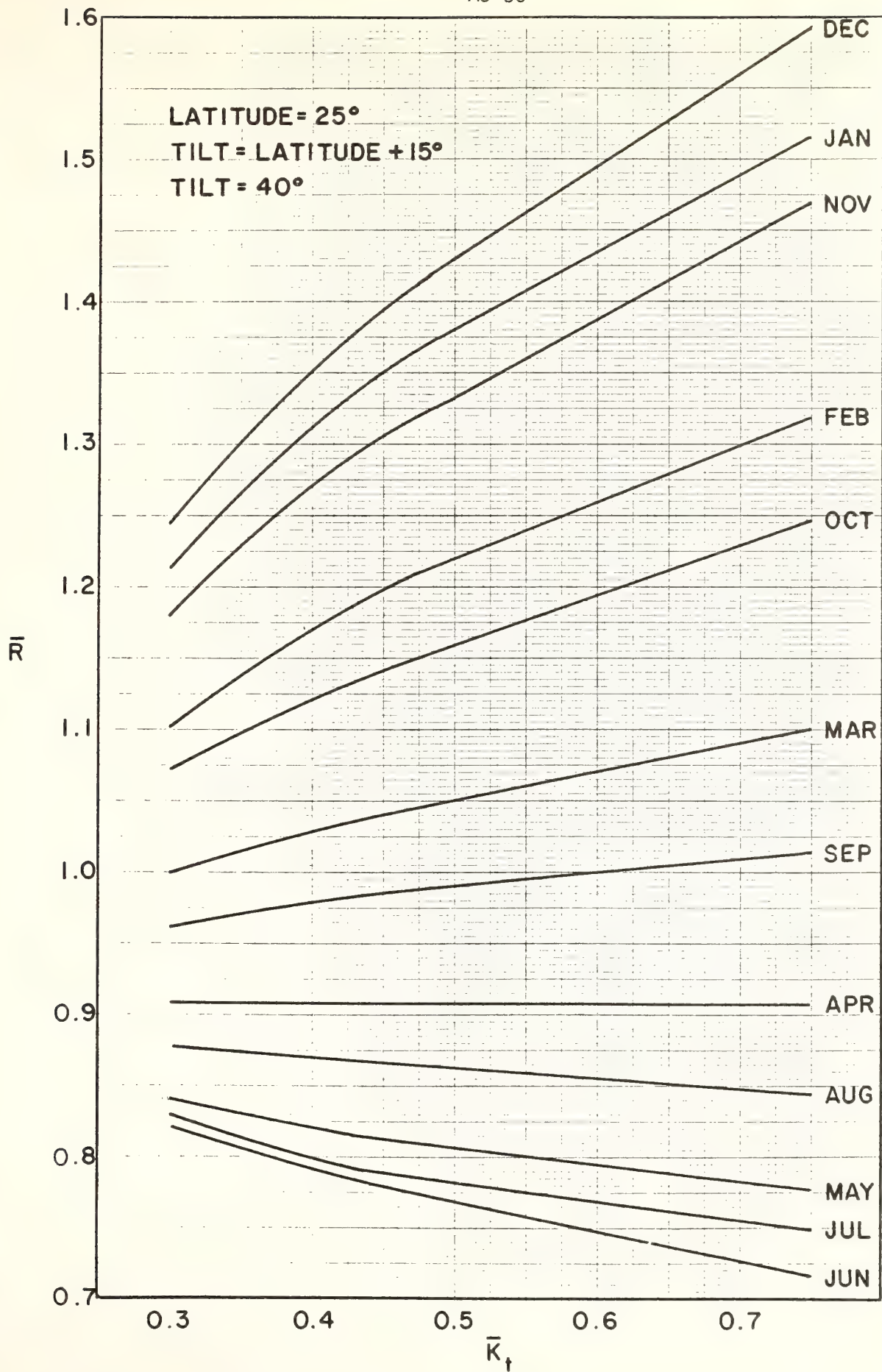


Figure A3-27. Tilt Correction Factor for 25° Latitude, 40° Tilt

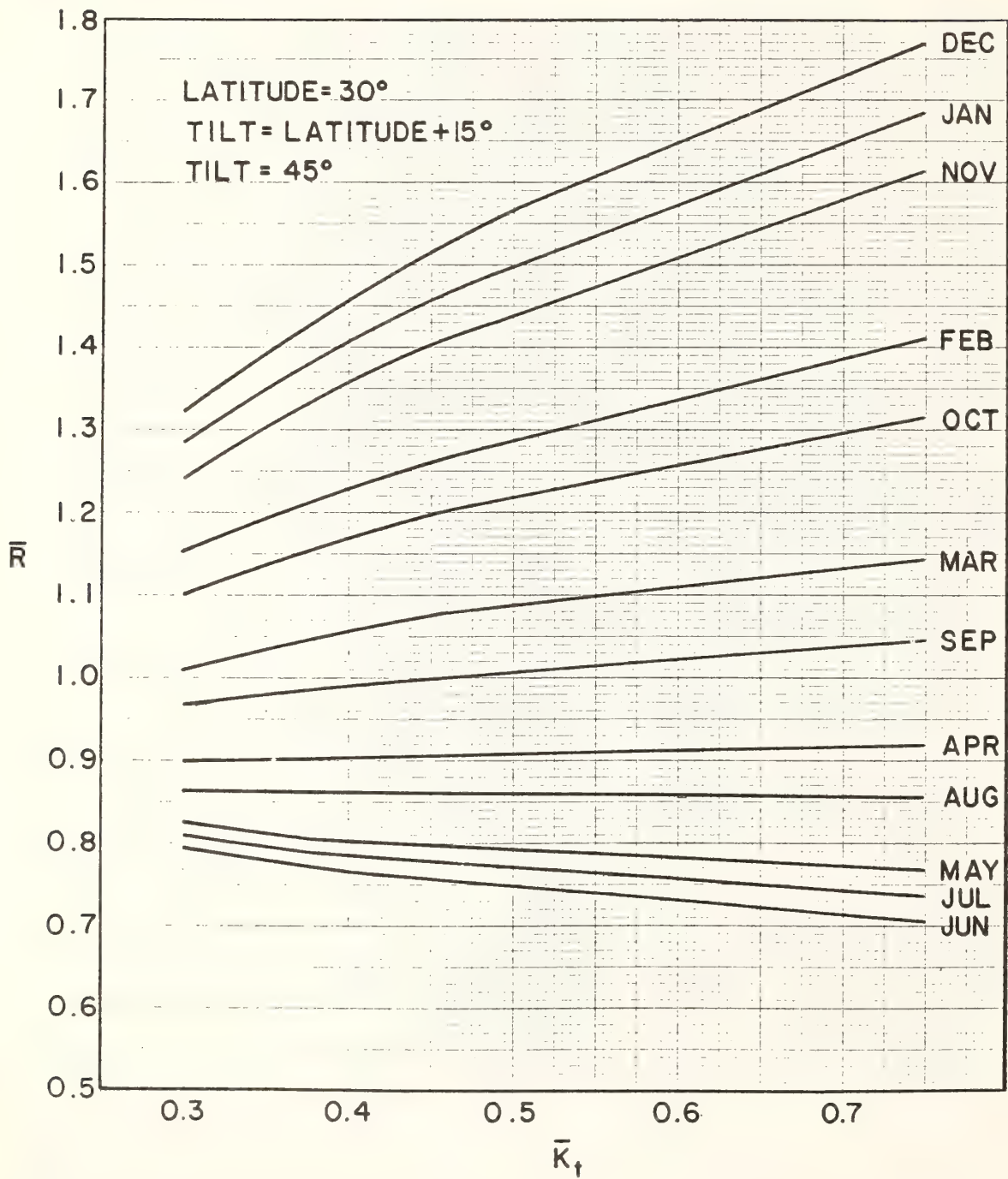
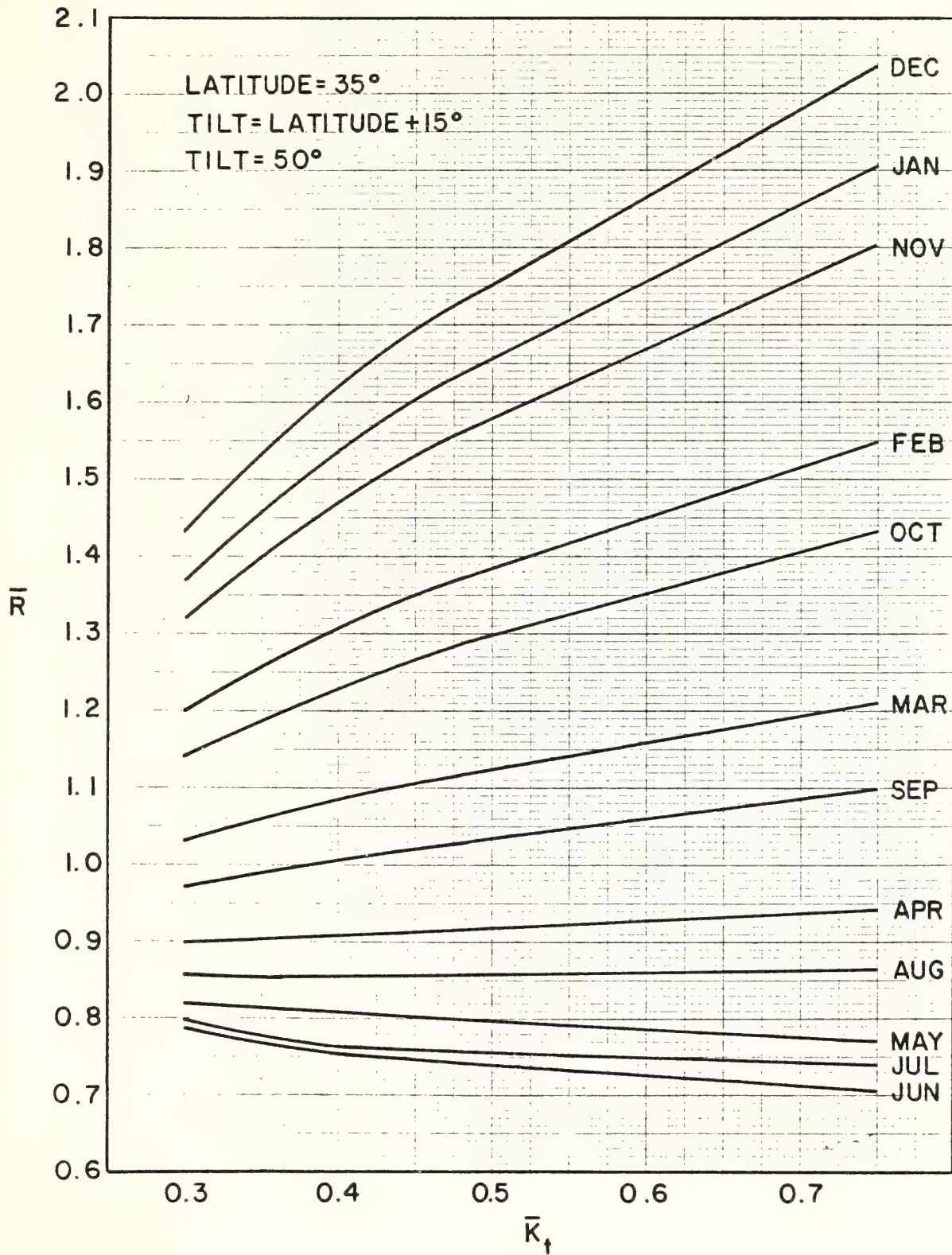


Figure A3-28. Tilt Correction Factor for 30° Latitude, 45° Tilt

Figure A3-29. Tilt Correction Factor for 35° Latitude, 50° Tilt

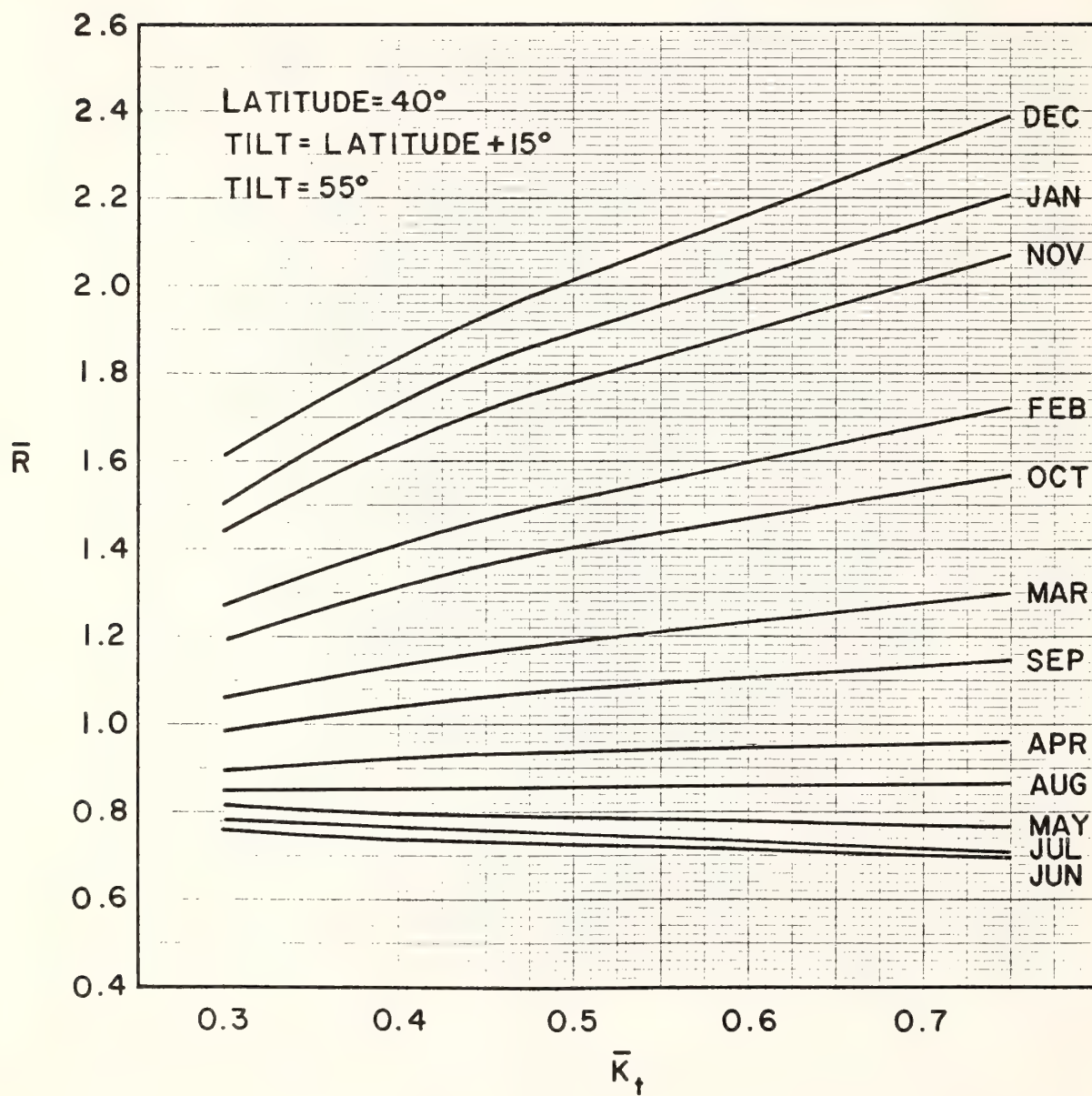


Figure A3-30. Tilt Correction Factor for 40° Latitude, 55° Tilt

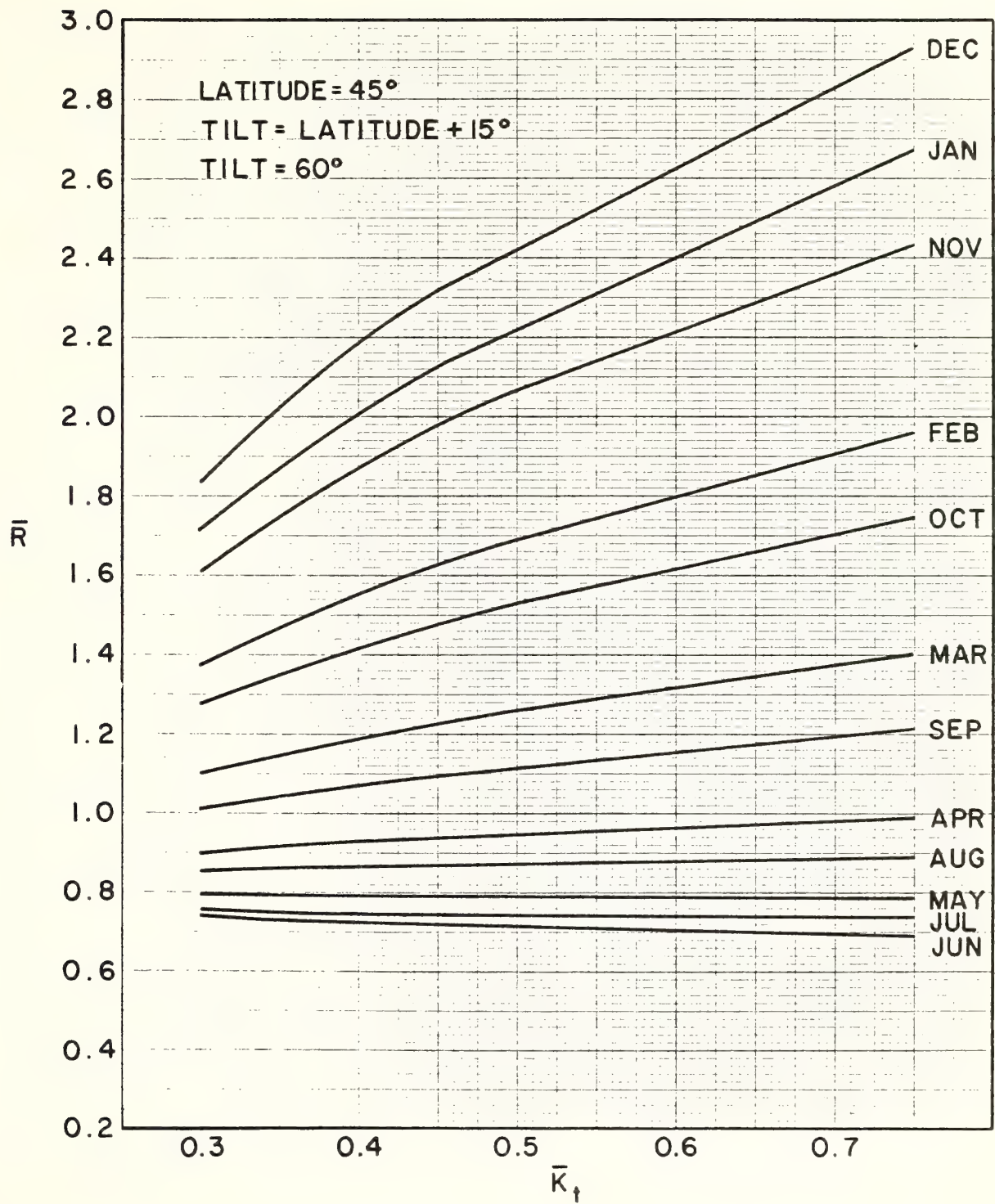


Figure A3-31. Tilt Correction Factor for 45° Latitude, 60° Tilt

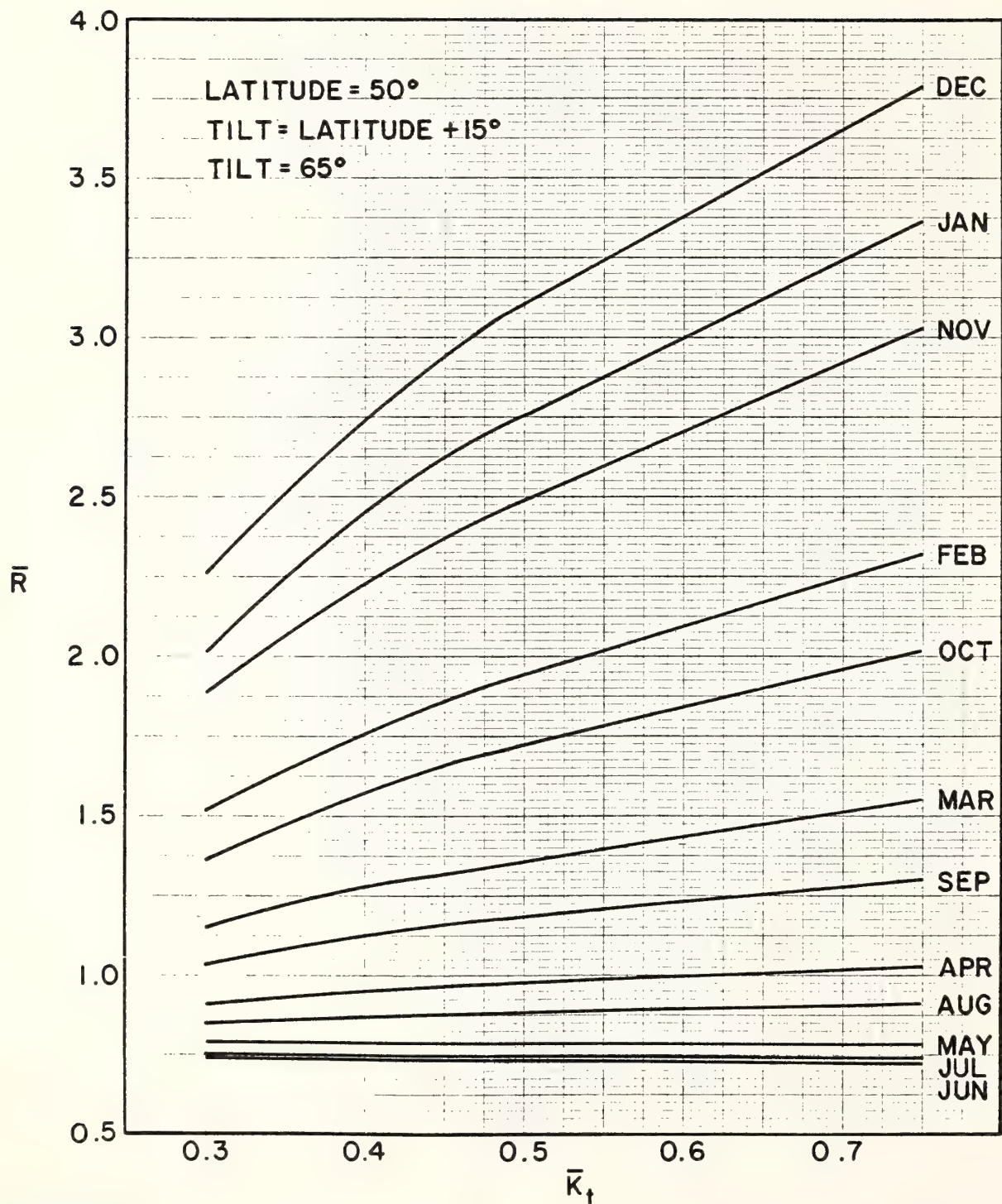


Figure A3-32. Tilt Correction Factor for 50° Latitude, 65° Tilt

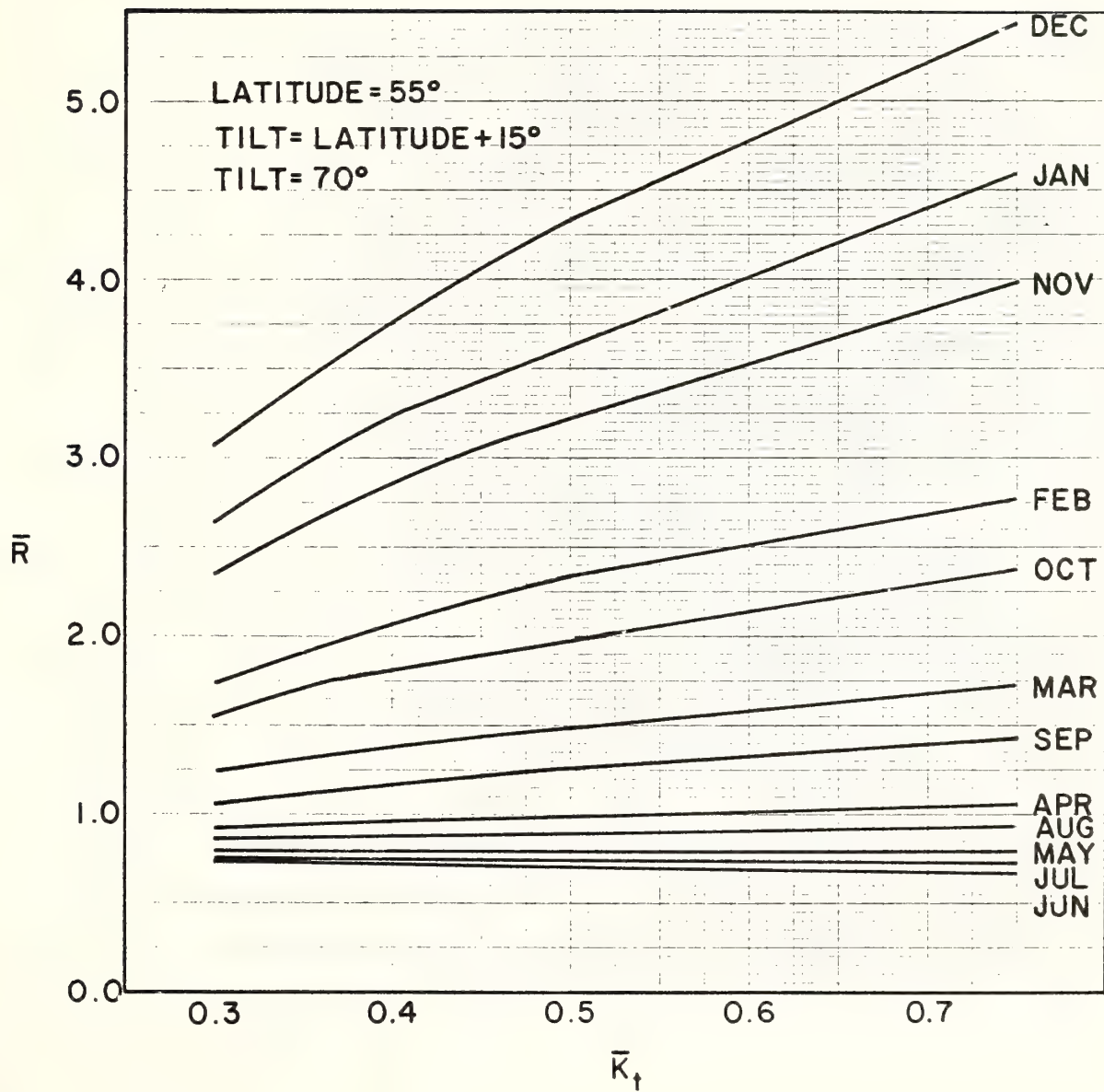


Figure A3-33. Tilt Correction Factor for 55° Latitude, 70° Tilt

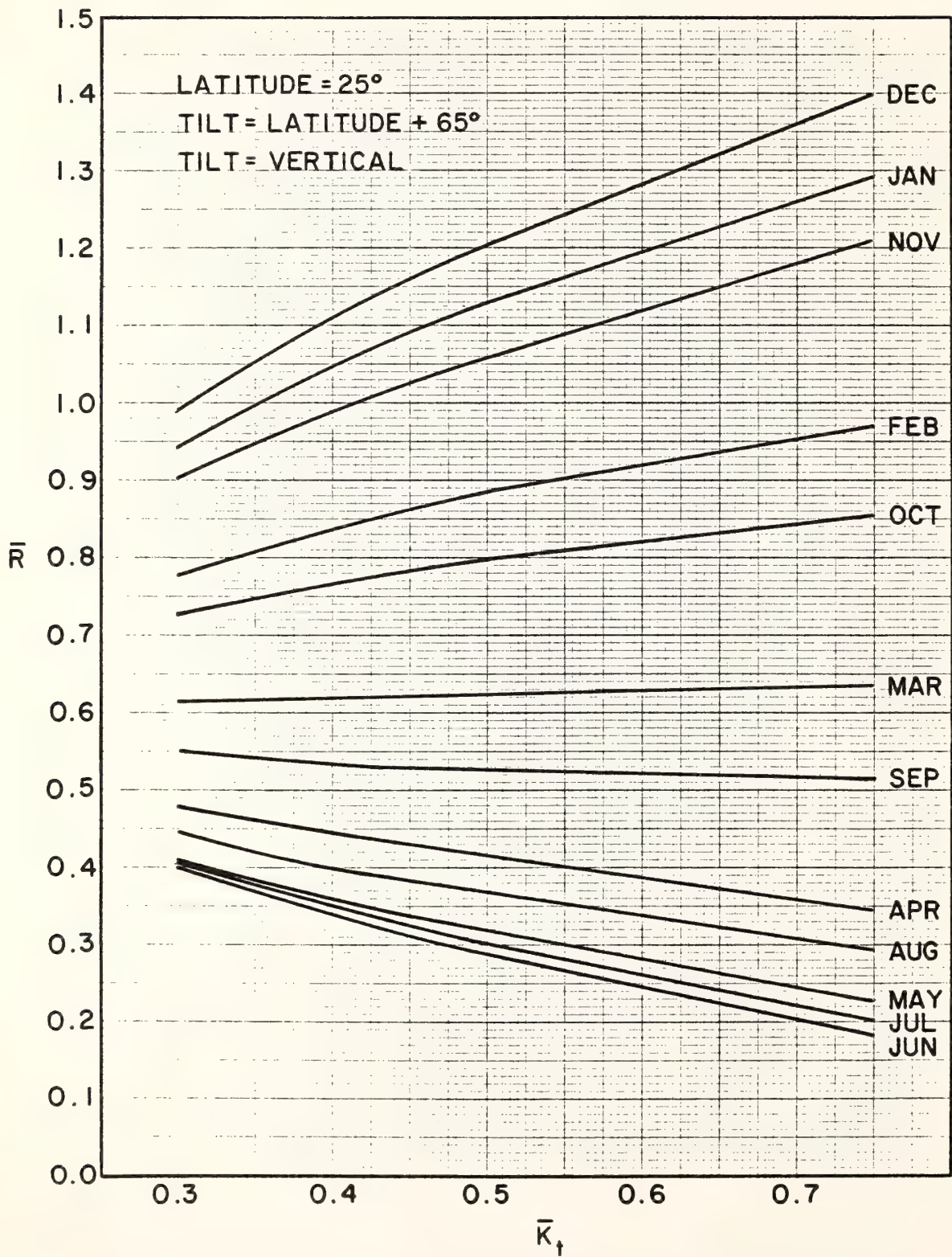


Figure A3-34. Tilt Correction Factor for 25° Latitude, Vertical Tilt

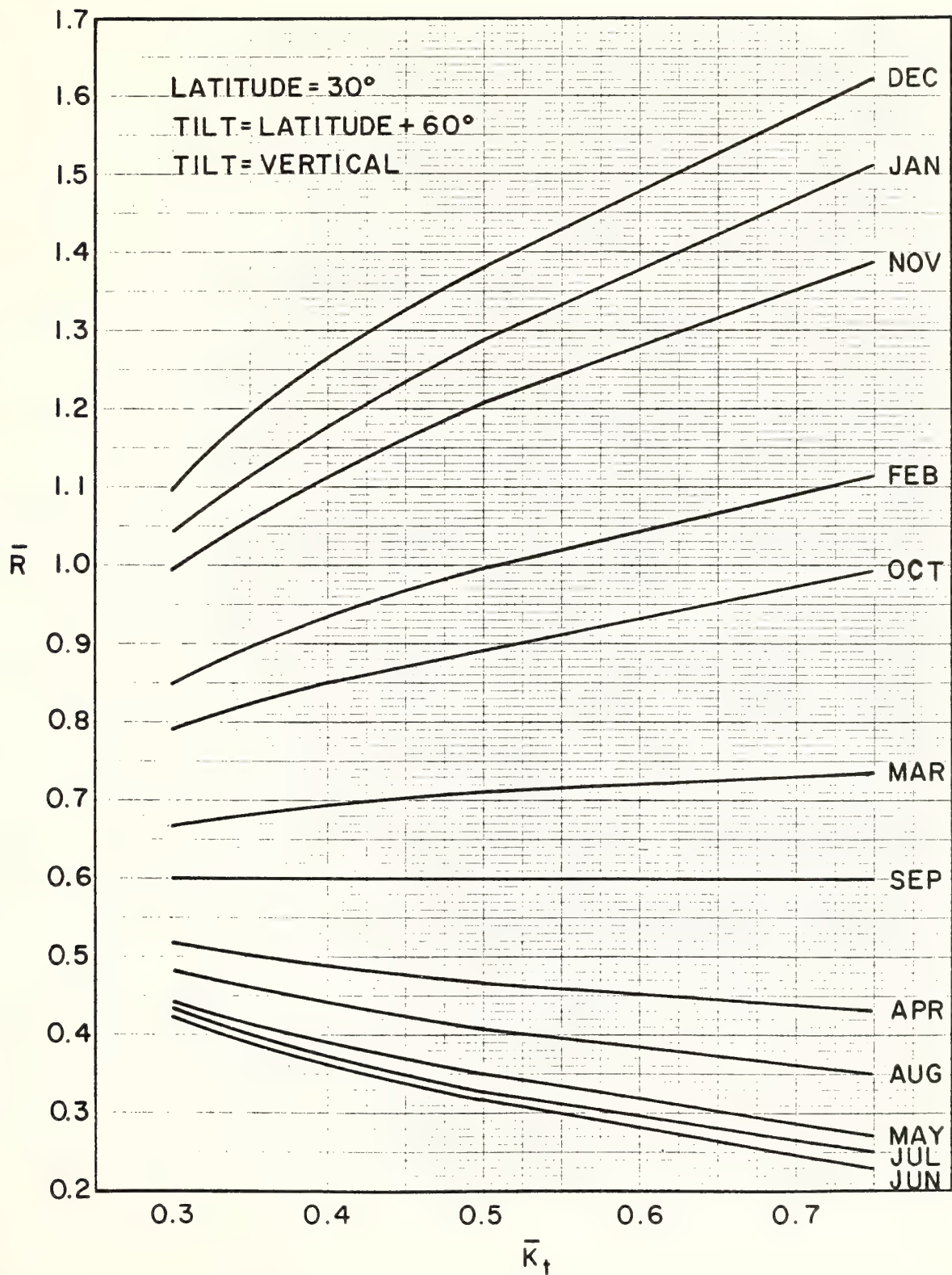


Figure A3-35. Tilt Correction Factor for 30° Latitude, Vertical Tilt

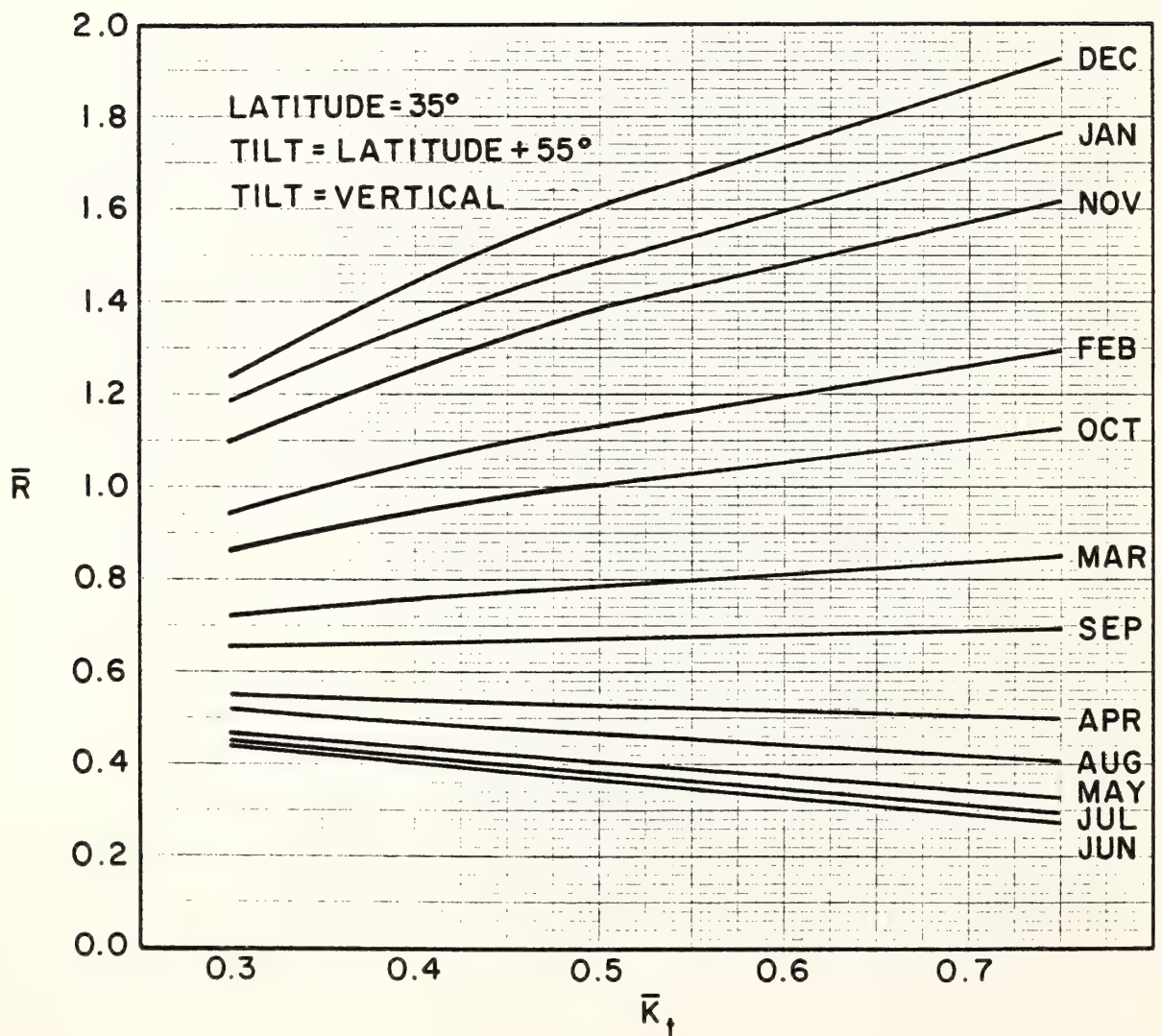


Figure A3-36. Tilt Correction Factor for 35° Latitude, Vertical Tilt

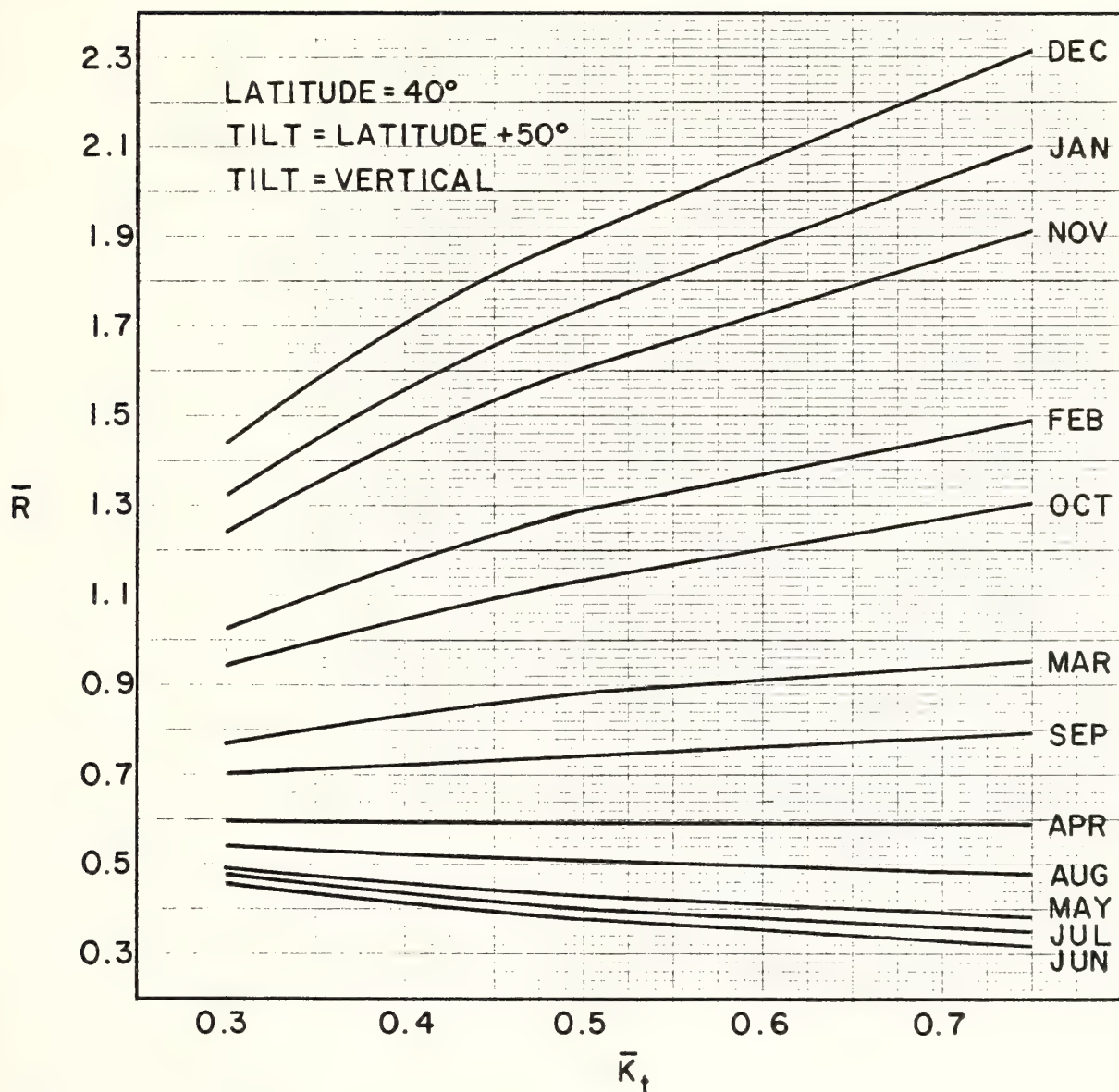


Figure A3-37. Tilt Correction Factor for 40° Latitude, Vertical Tilt

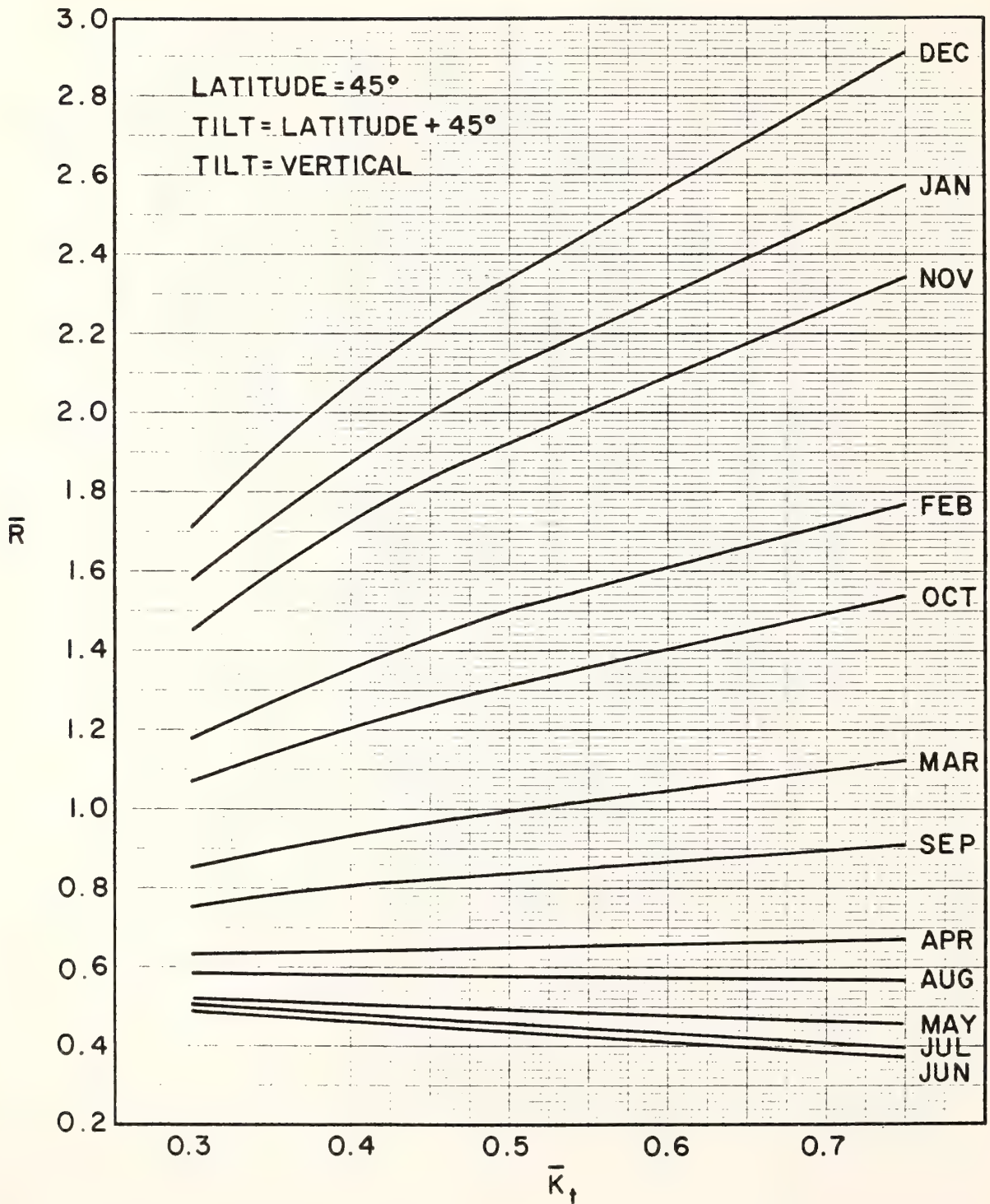


Figure A3-38. Tilt Correction Factor for 45° Latitude, Vertical Tilt

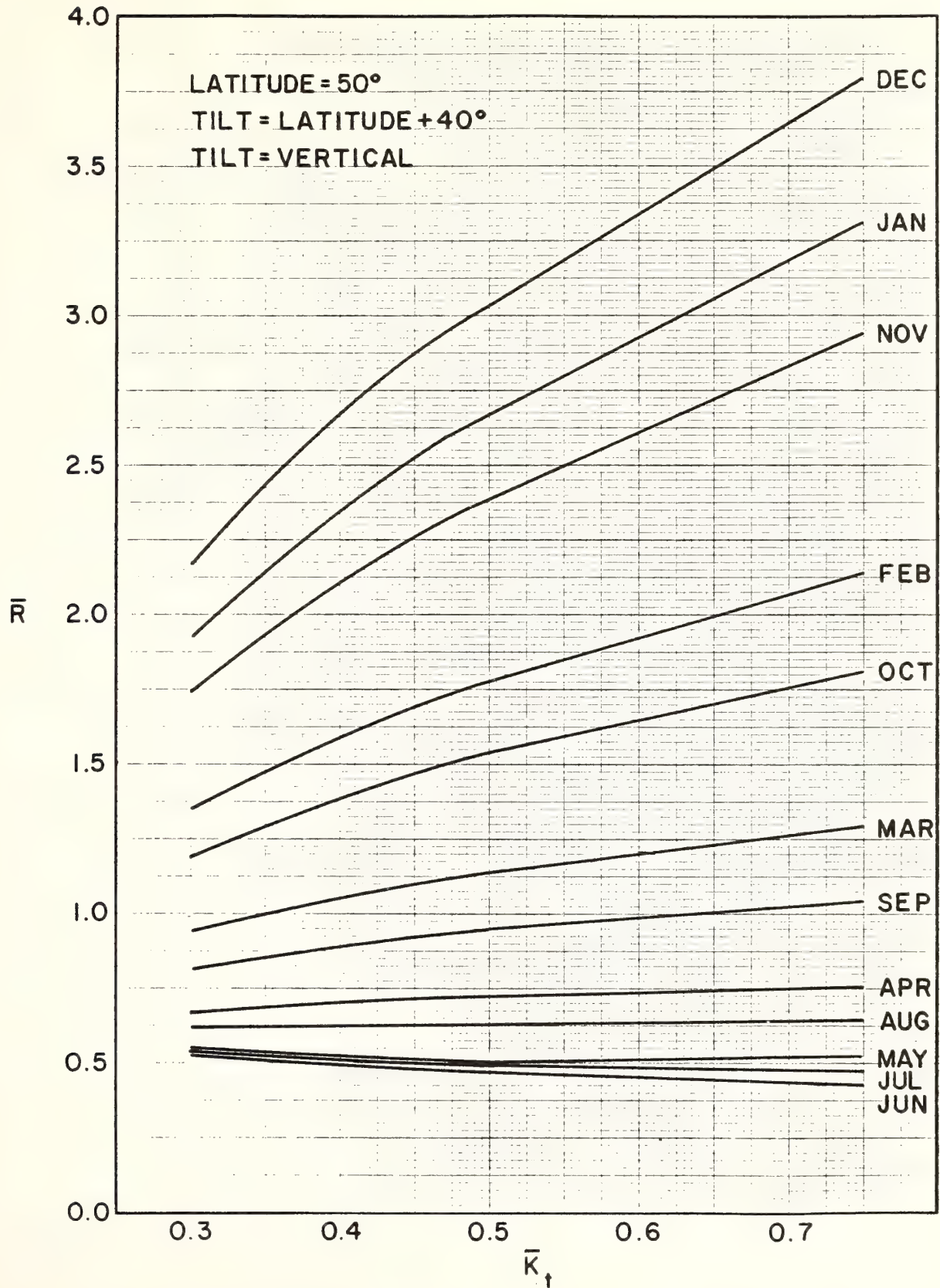


Figure A3-39. Tilt Correction Factor for 50° Latitude, Vertical Tilt

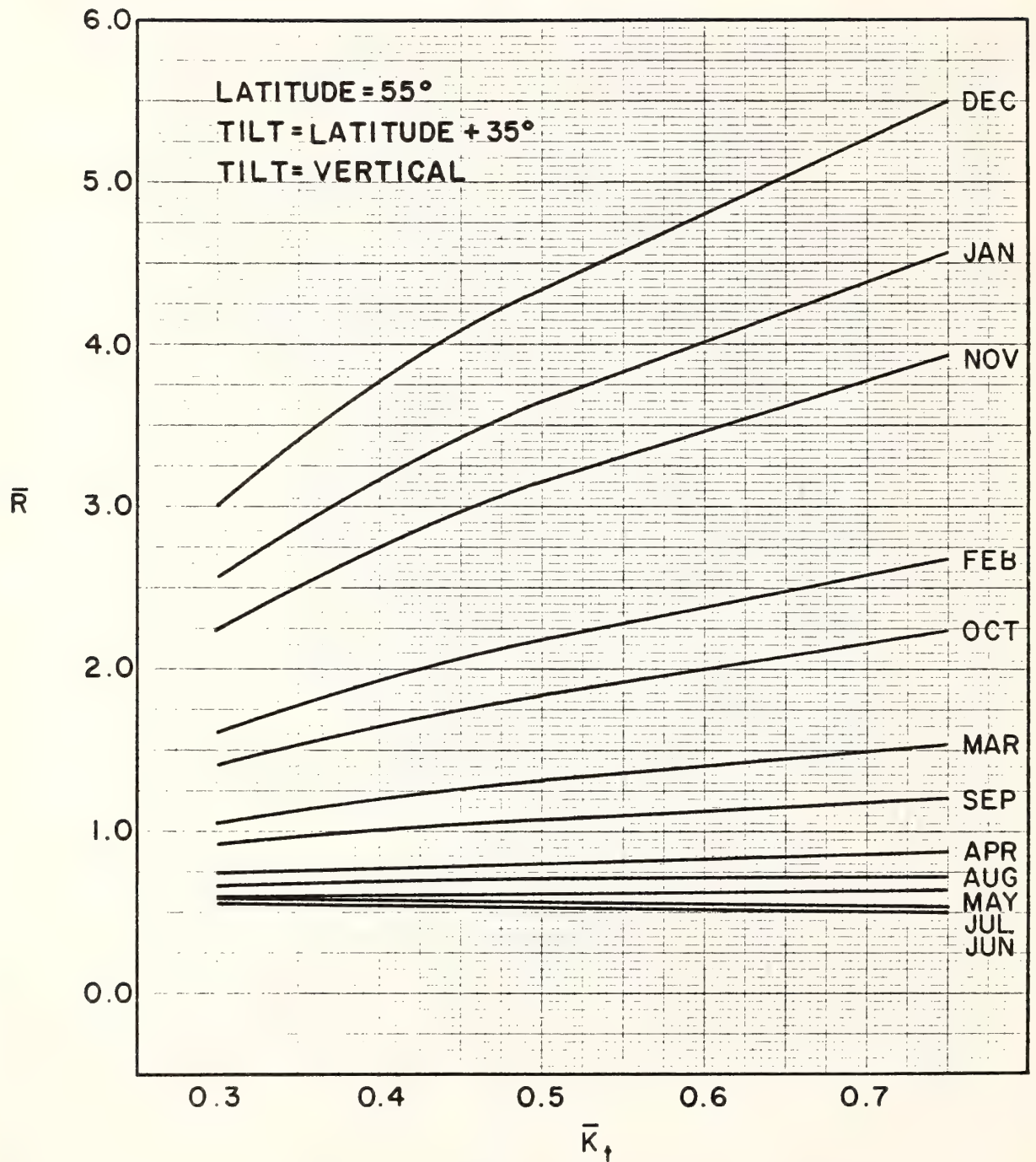


Figure A3-40. Tilt Correction Factor for 55° Latitude, Vertical Tilt

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 4

SOLAR COLLECTORS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO



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LIST OF SYMBOLS

A_C	Overall collector area or absorber area, ft^2
c_p	Specific heat of fluid, $\text{Btu}/(\text{lb}\cdot^\circ\text{F})$
C_C	Heat capacity flow rate = $\dot{m}c_p$, $\text{Btu}/(\text{hr}\cdot^\circ\text{F})$
F_R	Heat recovery factor, a correction factor in the collector performance equation when T_i is used instead of \bar{T}_p
I	Solar insolation, $\text{Btu}/(\text{hr}\cdot\text{ft}^2)$
I_o	Extraterrestrial radiation on a horizontal surface at the outer limits of earth's atmosphere, $\text{Btu}/(\text{hr}\cdot\text{ft}^2)$
I_T	Solar radiation on the outer cover of a tilted collector per unit area, $\text{Btu}/(\text{hr}\cdot\text{ft}^2)$
\dot{m}	Fluid mass flow rate, lb/hr
Q_u	Useful heat delivered by the collector, Btu/hr
T_a	Atmospheric temperature, $^\circ\text{F}$
T_i	Fluid temperature at the inlet to the collector, $^\circ\text{F}$
\bar{T}_p	Average temperature of the upper surface of the absorber plate, $^\circ\text{F}$
U_L	Overall heat loss coefficient from the collector per unit collector area or absorber area, $\text{Btu}/(\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F})$
α	Absorptance of solar radiation by the absorber plate, no dimension
η_C	Collector efficiency, useful heat delivered by the collector divided by the total solar radiation incident on the collector
τ	Transmittance of solar radiation through the covers, no dimensions
$(\tau\alpha)_n$	Transmittance absorptance product when beam radiation is normal to the plane of the collector cover

OBJECTIVES

At the end of this module, the trainee should be able to:

1. Identify and describe the purpose of each component of a solar collector.
2. Compare the performance of various collectors.
3. Describe methods of preventing corrosion and freezing of collectors.
4. Describe the use of various fluids in collectors.
5. Recognize effect of system design on collector performance.
6. Explain the factors contributing to solar collector durability.

INTRODUCTION

Collectors may be divided into two classes, liquid-heating and air-heating solar collectors. A common design for both types consists of an absorber plate, with black surface coating, contained in a metal frame box with one or more transparent covers above the absorber plate. The covers are transparent to incoming solar radiation and relatively opaque to outgoing (long-wave) radiation. The principal purpose of transparent covers is to reduce heat losses. Insulation is placed beneath the absorber plate to reduce heat losses through the back of the collector. A special collector design uses a vacuum jacket around the absorber to reduce conduction and convection heat losses.

Nearly all practical systems for solar space heating and hot water heating involve flat-plate collectors. They are easy to install, require no moving parts, and have reasonable prospects for reliable and durable operation. Some types of concentrating collectors are being tested in experimental situations, but their cost is much higher than the flat-plate types, and there is not sufficient information available at the present time to evaluate the practicality of concentrating collector systems.

TYPICAL SOLAR COLLECTORS

TYPICAL LIQUID COLLECTOR

A partially sectioned diagram of a typical flat-plate liquid solar collector is shown in Figure 4-1. The drawing shows a commercially manufactured collector comprising a glass-covered metal box containing an absorber plate to which an array of tubes is attached and beneath which insulation is provided. A liquid is pumped through the collector tubes and manifolds for heating. Typical collector dimensions are 6.5 feet by 3.0 feet. The space between glass covers is about one-half inch and the inner glass cover is about one inch above the absorber plate. Two to four inches of insulation such as heat-resistant fibrous glass are commonly used below the absorber plate. Metal is probably the best material for absorber plates, and good thermal contact is required between the absorber plate and the tubes through which the liquid is transported. Volumetric flow rate is typically $0.02 \text{ gal}/(\text{min} \cdot \text{ft}^2)$ of collector surface area.

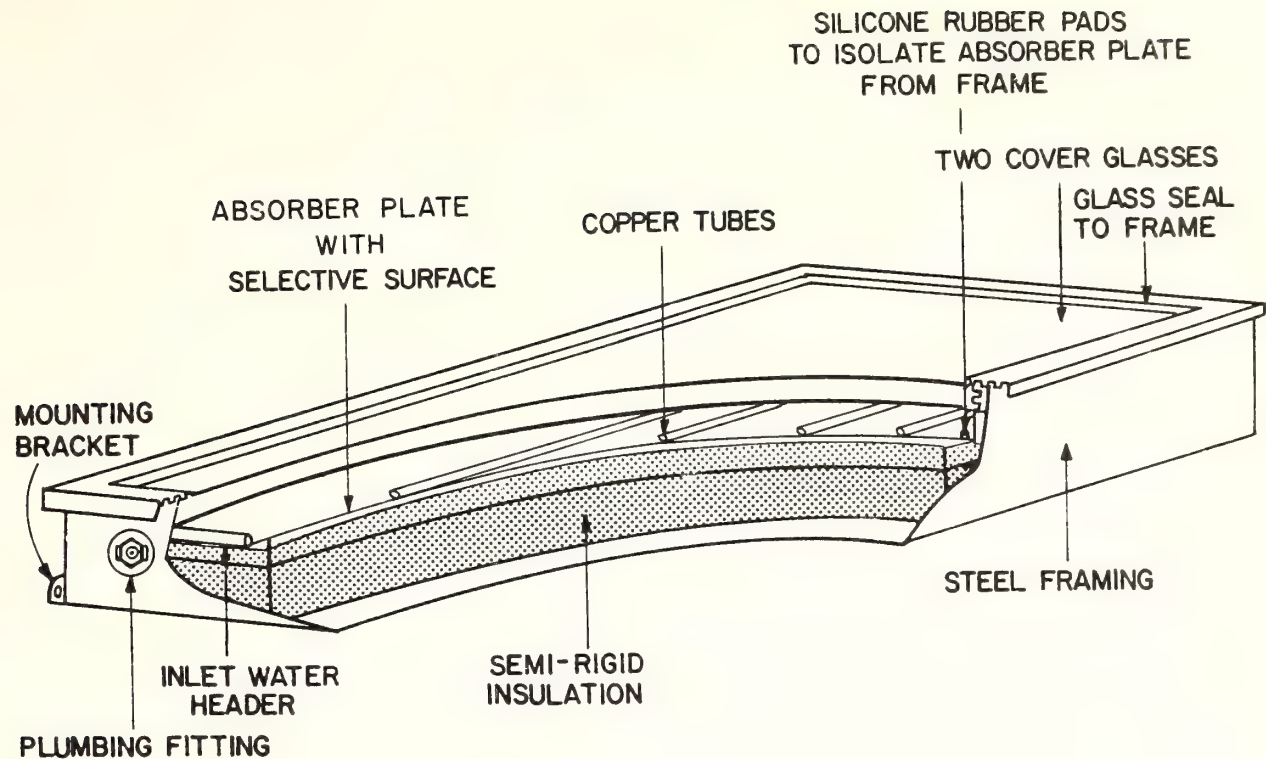


Figure 4-1. Liquid-Heating Solar Collector

TYPICAL AIR COLLECTOR

A sketch of a typical air-heating solar collector is shown in Figure 4-2. The principal difference between air- and liquid-heating collectors is the size and configuration of the fluid conduits. The figure shows three wide air passages directly beneath the absorber plate. Air therefore flows in contact with nearly the entire absorber surface, through channels about one-half inch high, at a sufficient velocity for effective heat transfer. The design shown also has an internal manifold for air distribution to the three channels of the collector panel. Volumetric flow rate is typically 2 cfm/ft^2 of collector surface area.

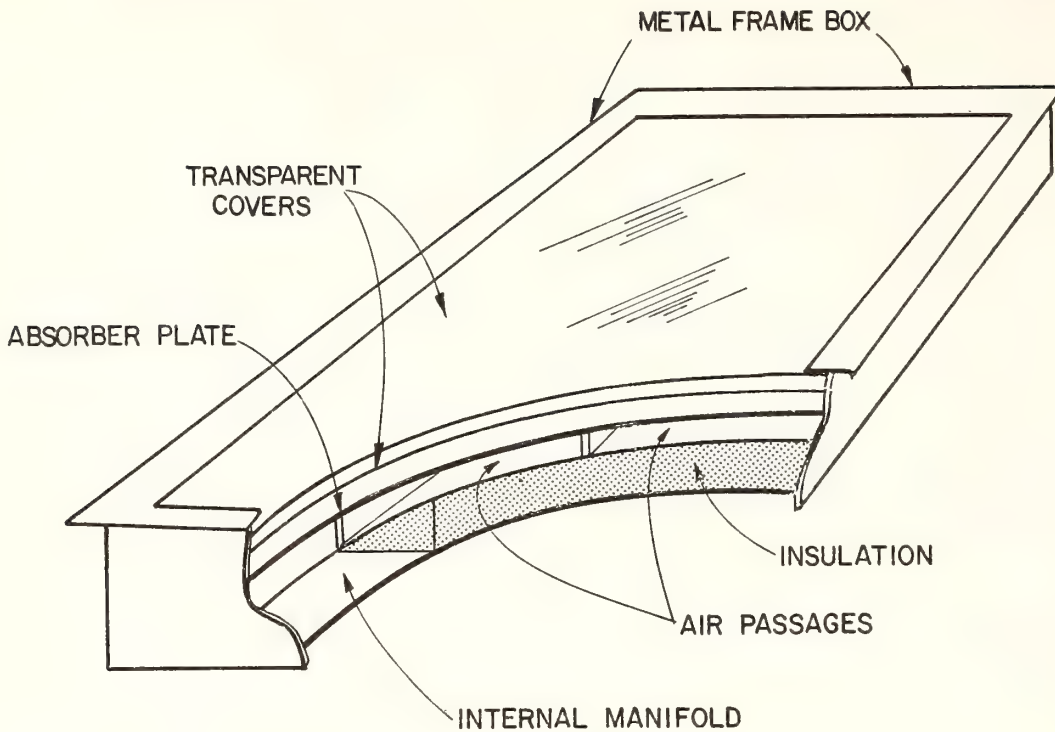


Figure 4-2. Air-Heating Solar Collector

BASIC PRINCIPLES

A solar collector is a device for converting the energy in solar radiation to heat in a fluid. This conversion is accomplished by absorbing the solar radiation on a broad, thin metal surface which is in contact with a stream of liquid or gas. Absorption of solar energy causes the temperature of the metal surface to rise so that the temperature of the fluid increases as it moves past the surface.

Under steady conditions, the useful heat delivered by the solar collector is equal to the energy absorbed in the metal surface minus the heat losses from that surface directly and indirectly to the surroundings. This principle can be stated in the relationship:

$$\begin{array}{c} \text{Useful} \\ \text{heat} \\ \text{gain} \end{array} = \begin{array}{c} \text{Area} \\ \text{of} \\ \text{absorber} \end{array} \left[\begin{array}{c} \text{Solar} \\ \text{radiation} \\ \text{absorbed} \end{array} - \begin{array}{c} \text{Heat} \\ \text{losses} \end{array} \right]$$

or rewriting in equation form,

$$Q_u = A_c [I_T \tau \alpha - U_L (\bar{T}_p - T_a)] \quad (4-1)$$

where

Q_u is useful heat delivered by the collector, Btu/hr

A_c is total collector area, ft^2

I_T is the solar energy received on the upper surface of the sloping collector per unit area, $\text{Btu}/(\text{hr} \cdot \text{ft}^2)$

τ is fraction of the incoming solar radiation which reaches the absorbing surface, no dimensions

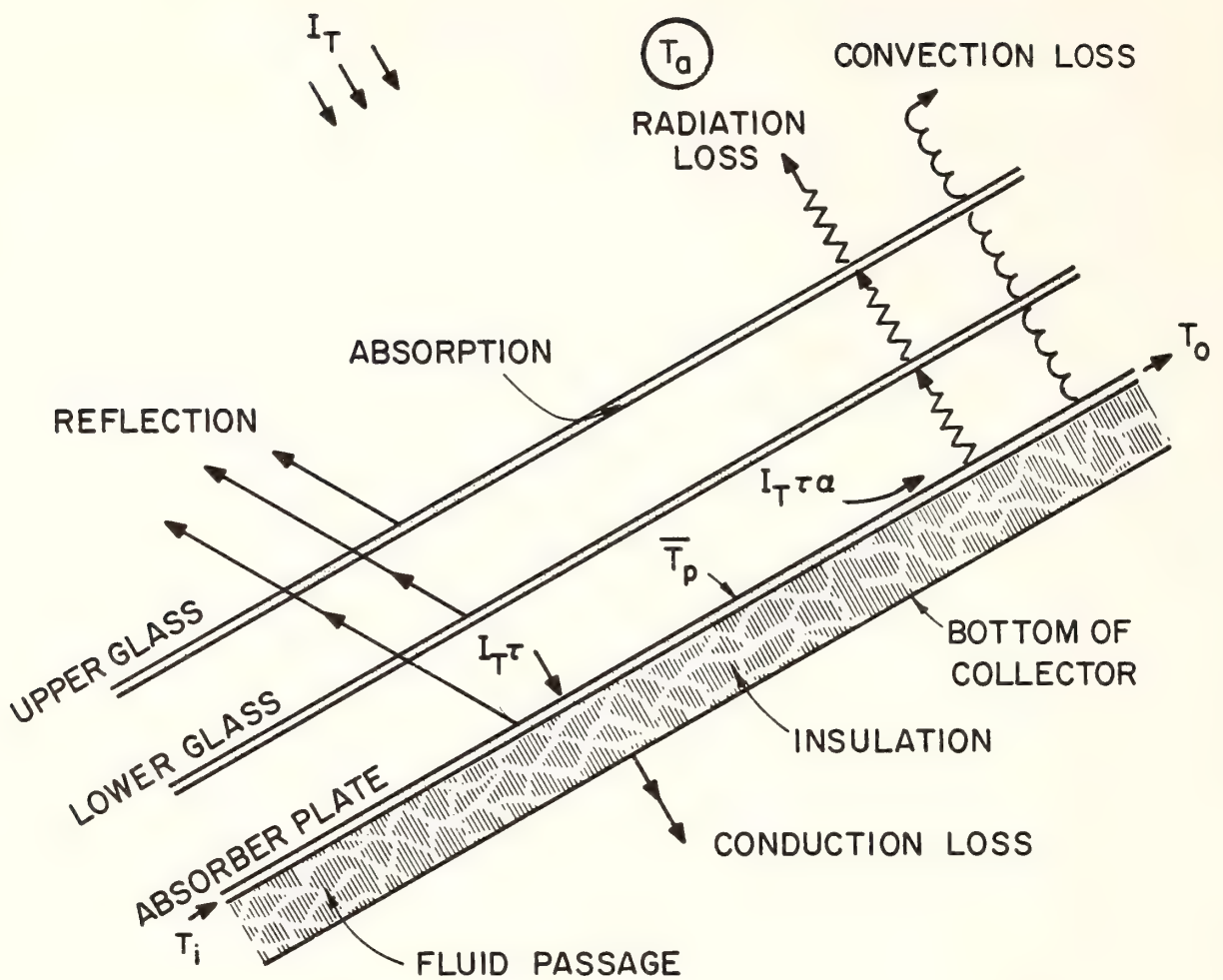
α is fraction of the solar energy reaching the surface which is absorbed, absorptivity, no dimensions

U_L is the overall heat loss coefficient, $\text{Btu}/(\text{hr} \cdot \text{ft}^2 \cdot ^\circ\text{F})$ transferred to the surroundings from each square foot of exposed collector surface per degree difference between average collector surface temperature and the surrounding air temperature

\bar{T}_p is average temperature of the upper surface of the absorber plate, $^\circ\text{F}$

T_a is atmospheric temperature, $^\circ\text{F}$

A diagrammatic representation of the terms in this relationship is shown in Figure 4-3.



$$\text{ABSORBED ENERGY} = A_c I_T \tau \alpha$$

$$\text{EFFECTIVE HEAT LOSS} = A_c U_L (\bar{T}_p - \bar{T}_a)$$

Figure 4-3. Definition Sketch for Equation 4-1

HEAT LOSSES FROM COLLECTOR

In order that the performance of a solar collector can be as high as economically practical, the design and operating factors which can maximize the value of the first term on the right-hand side of Equation (4-1) and can minimize the value of the second term are selected. In other words, the greater the energy absorption in the metal surface and the lower the heat loss from that surface, the higher will be the useful recovery. If a bare metal plate serves as the collector, and with typical values of 2 to 10 Btu/(hr·ft²·°F) for the coefficient of heat transfer to the atmosphere, U_L , the rates of heat loss will be so large that an absorber plate temperature 25 to 50 degrees above atmospheric temperature would be the maximum achievable under typical full solar radiation of 300 Btu/(hr·ft²). Under these conditions no useful heat would be delivered from the collector because the heat loss would be equal to the solar heat absorbed, leaving nothing for useful delivery.

To reduce the rate of heat loss occurring by radiation and convection, one or more transparent surfaces, such as glass, can be placed above the metal surface. The glass will transmit as much as 90 percent of the incident solar radiation and it will greatly reduce the heat loss coefficient, U_L . The reduction in U_L is due to the suppression of convection losses from the absorber plate by the relatively stagnant air layer between the absorber plate and the glass, and by intercepting the long-wave, thermal radiation emitted by the hot metal surface because glass is opaque to the long-wave radiation. The heat loss coefficient can be reduced to 1 or 2 Btu/(hr·ft²·°F) by the use of one glass cover. Similar benefits can be realized by use of certain transparent plastic materials.

Further reduction in the heat loss coefficient can be realized by using a second transparent surface with an air space between the two surfaces. Two relatively stagnant air barriers to convection loss are then present, as well as two surfaces impeding radiation loss. Coefficients in the range of $0.7 \text{ Btu}/(\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F})$ are typically then obtained.

Radiation losses can be reduced by other techniques, such as by reducing the radiation-emitting characteristics of the heat absorbing surface. This measure is discussed in the section pertaining to the solar radiation absorbing characteristics of the surface. Thermal radiation emitted by the absorber plate may also be reduced by reflecting it downward from the lower glass cover by applying an infrared-reflecting coating on the glass. An optically transparent, very thin layer of tin oxide or indium oxide deposited on the glass will reduce radiation loss by reflecting it back to the absorber plate. This coating absorbs a small fraction of the solar radiation, however, so the reduced thermal loss is largely off-set by reduced solar energy input to the absorber plate.

Significant losses can occur from the side and back of the collector unless insulation is used. It is advisable to use a high-temperature insulation adjacent to the back side of the absorber plate layered with a lower temperature insulation to provide the required resistance to heat flow. The total R value of the insulation should be at least 10 for medium-temperature flat-plate collectors.

A transparent honeycomb of thin plastic film can also suppress radiation loss if interposed between the absorber plate and the lower glass cover. Convection loss suppression also can be achieved, leading to improvement of overall efficiency. Low- to moderate-priced plastic

film does not appear to have sufficient resistance to damage by high collector-plate temperatures, however, so this technique has not been commercially utilized.

The foregoing discussion has been concerned with methods for reducing U_L , the heat loss coefficient, to the lowest practical level. By so doing, the total heat loss is minimized and collector efficiency is increased. It is evident from Equation (4-1) that losses also decrease as the difference between plate temperature and air temperature decreases. The ambient (outside) air temperature is an uncontrollable factor, of course, but the fact that it varies with time and with geographic location means that collector efficiency will also be dependent upon these factors. It is clear, also, that a collector will be more efficient at lower plate temperatures than at high temperatures. Plate temperature is dependent largely on the way the collector is operated, that is, by the temperature of the fluid being circulated in contact with the plate, the rate of fluid circulation, and the type of fluid. Fluid temperature depends on conditions elsewhere in the system, whereas the other factors depend on the collector design and the operating conditions.

SOLAR ENERGY ABSORPTION

In Equation (4-1), the first term is the solar energy absorbed in the absorbing surface, which depends upon the solar energy incident on the tilted surface of the collector and is affected by collector orientation, as outlined in Module 3. This climatic variable can be measured or derived from tables of averages, and if not already converted, can be calculated for the proper collector position.

The transmissivity of the glass, τ , is a function of the quality of the glass and the angle at which the solar radiation reaches the glass. At normal incidence (solar beam perpendicular to the glass surface), one sheet of ordinary window glass reflects about eight percent of the solar radiation. Two sheets of glass with air space between reflect about 15 percent. Impurities in the glass, principally iron, result in some radiation absorption; typical glasses 1/8 inch in thickness absorb one to five percent per sheet. Glass with reasonably low iron content may absorb about two percent per sheet, so at normal incidence, the total transmission of two sheets of glass can be approximately 80 percent. The value of τ is, therefore, 0.8.

Because the beam radiation from the sun strikes the collector at an angle which varies throughout the day, as well as seasonally, a weighted mean transmissivity is somewhat lower than this normal-incidence value. Precise calculations can be made, but a satisfactory approximation for a single-glazed collector can be based on a 10 percent average reflection loss and a suitable absorption loss dependent on glass quality. Assuming two percent absorption, an average transmissivity, τ , of a single sheet could be about 0.88. In a double-glazed collector, an effective transmission coefficient of 0.78 could be obtained with good quality glass.

If plastics are used for the transparent surfaces, transmission coefficients could be appreciably different, depending upon the characteristics of the plastics. Some have transmissivities moderately higher than glass, whereas others show lower values.

Methods for reducing the reflectivity of glass surfaces have been developed. Metallic films formed by vapor deposition are commonly used

as lens coatings in photographic equipment. These interference layers are too costly for use in solar collectors. Another process involves a delicate etching of the glass surface by acid treatment, producing essentially a slightly porous silica surface. Solar reflectivities as low as 1 to 2 percent can be obtained under carefully controlled conditions. Total transmissivity of a double-glazed collector can thereby be increased to values above 90 percent. The cost-effectiveness of this substantial improvement in performance has yet to be established.

The solar absorptivity of the radiation-receiving surface, α , is dependent on the optical property of the materials exposed to solar radiation. Surfaces which appear black to the eye have high absorptivity for the visible portion of the solar spectrum, and usually also are good absorbers for the infrared portion of the solar radiation. Carbon black, numerous oxides, and most black paints have absorptivities above 0.95, that is, they absorb 95 percent of the solar radiation reaching the surface. The remainder of the solar radiation is reflected upwards through the glazing. The overall efficiency of the collector is strongly dependent on the absorptivity of this surface.

The most common types of absorber surfaces are heat-resistant black paints, usually applied by spraying, followed by curing with heat to eliminate solvents and to secure permanence. These surfaces must be capable of prolonged exposure to temperatures of 300°F to 400°F in double-glazed collectors, without appreciable deterioration or outgassing. In a recently developed solar air collector, sheet steel coated with black porcelain enamel (applied to the steel as a sprayed-on frit and fused to the surface in a furnace) is achieving successful application.

SELECTIVE SURFACES

Most surfaces that are good absorbers for solar radiation are also good radiators of heat. If, for example, a surface has an absorptivity of 0.95 for solar radiation, it will normally radiate heat at a rate about 95 percent of that of a "perfect" radiator. Certain combinations of surfaces, known as selective surfaces, are capable of absorbing solar radiation effectively, while at the same time radiating heat at a low rate. Most selective surfaces are composed of a very thin black metallic oxide on a bright metal base. The black oxide coating is thick enough to act as a good absorber, with an absorptivity as high as 0.95, but it is essentially transparent to long-wave thermal radiation emitted by an object at a temperature of several hundred degrees F. Because bright metals have low emissivity for thermal radiation, that is, are poor heat radiators, and the thin oxide coating is transparent to such radiation, the combination is a poor heat radiator. As a result, the radiation loss from this type of surface is considerably lower than from a conventional, non-selective surface. The overall heat loss coefficient for the collector, U_L , has a lower value when a selective surface is used.

The most successful and stable selective surface developed to date is made by electroplating a layer of nickel on the absorber plate, then electrodepositing an extremely thin layer of chromium oxide on the nickel substrate. Nickel oxide coatings have also been used, but they are less resistant to damage from moisture. Coatings of copper oxide on bright copper and nickel have similar properties, but temperature stability is limited. The most effective selective surfaces have solar absorptivities near 0.95 and thermal emissivities near 0.1.

COLLECTOR PERFORMANCE

CONVENIENT PERFORMANCE EQUATION

Having now recognized the principal design factors affecting collector performance, specifically those related to heat loss control and those involving the absorption of solar radiation, we now can see from Equation (4-1) that if the numerical values of all the terms are known, the rate of useful heat recovery, Q_u , can be calculated. In addition to the design characteristics of the collector discussed above, the three operating conditions, solar radiation, average absorber-plate temperature, and ambient temperature must be known. With the exception of plate temperature, these terms can readily be measured or obtained from tables or charts. Absorber-plate temperature, however, is seldom known, nor can it be easily determined. It is affected by the other collector operating conditions and, most critically, by the temperature of the fluid being supplied to the collector.

In an operating system comprising collector, storage, and space being heated, the temperature of the fluid in storage can be calculated or assumed until confirmed. This fluid is supplied to the collector and strongly controls the absorber-plate temperature in Equation (4-1). In a typical liquid collector, average plate temperatures usually are 10 to 20 degrees above inlet liquid temperature and, in air collectors, the temperature difference is 30 to 50 degrees. As a convenience, therefore, Equation (4-1) can be modified by substituting inlet fluid temperature for the average plate temperature, if a correction factor is applied to the resulting useful heat determination. The resulting equation is

$$Q_u = F_R A_c [I_T \tau \alpha - U_L (T_i - T_a)] \quad (4-2)$$

where

T_i is the temperature of the fluid entering the collector

F_R is a correction factor or "heat recovery factor", having a value between 0 and 1.0, such that the useful heat recovery calculated by Equation (4-2) is equal to that calculated by Equation (4-1).

HEAT RECOVERY FACTOR

The heat recovery factor, F_R , can be interpreted as the ratio of the heat actually recovered to that which would be recovered if the collector plate were operating at a temperature equal to that of the entering fluid. This temperature equality would theoretically be possible if the fluid were circulated at such a high rate through the collector that there would be a negligible rise in the temperature of the fluid passing through the collector, and the heat transfer coefficient were so high that the temperature difference between the absorber surface and the fluid would be negligible.

In Equation (4-2), the temperature of the inlet fluid is dependent on the characteristics of the complete solar heating system and the heat demand of the building. F_R , however, is affected only by the collector characteristics and the fluid flow rate through the collector. As indicated above, the numerical value of F_R would be 1.0 if the entering fluid temperature and the average plate temperature were the same.

COLLECTOR TEMPERATURE PATTERNS

The better the heat transfer coefficient between the metal plate and the fluid, the more nearly the fluid temperature will approach the plate temperature at any one position in the collector, hence the higher will be the value of F_R . Similarly, the greater the fluid circulation rate, the smaller will be the temperature change from inlet to outlet and the closer will be the inlet fluid temperature to the average plate temperature. Figure 4-4 shows a typical temperature pattern in a solar heater being supplied with liquid at 130 degrees. Liquid leaves the collector at about 150 degrees, the collector-plate temperature is about 10 degrees above the liquid temperature throughout the collector, and the average plate temperature is about 150 degrees. If typical values of the collector parameters are substituted in Equations (4-1) and (4-2), it will be found that using 130°F as inlet fluid temperature in Equation (4-2) instead of 150°F as the average plate temperature in Equation (4-1) would necessitate use of a heat recovery factor, F_R , of about 0.9 to obtain the correct value of Q_u . If the coefficient of heat transfer between the collector plate and the liquid is lower, or if a lower fluid circulation rate is used, the value of F_R would be slightly less.

A temperature pattern in a typical air-heating collector operating with an air supply from the space being heated or from the cold end of a pebble-bed storage unit at 70°F is also shown in Figure 4-4. Full sun and a practical air circulation rate of about 2 cfm per square foot of collector are assumed in the example. An air temperature rise of about 60 to 80 degrees would occur under these conditions, which is much higher than in the liquid case because of the lower specific heat for air. The mass flow rate is about the same as that of the liquid

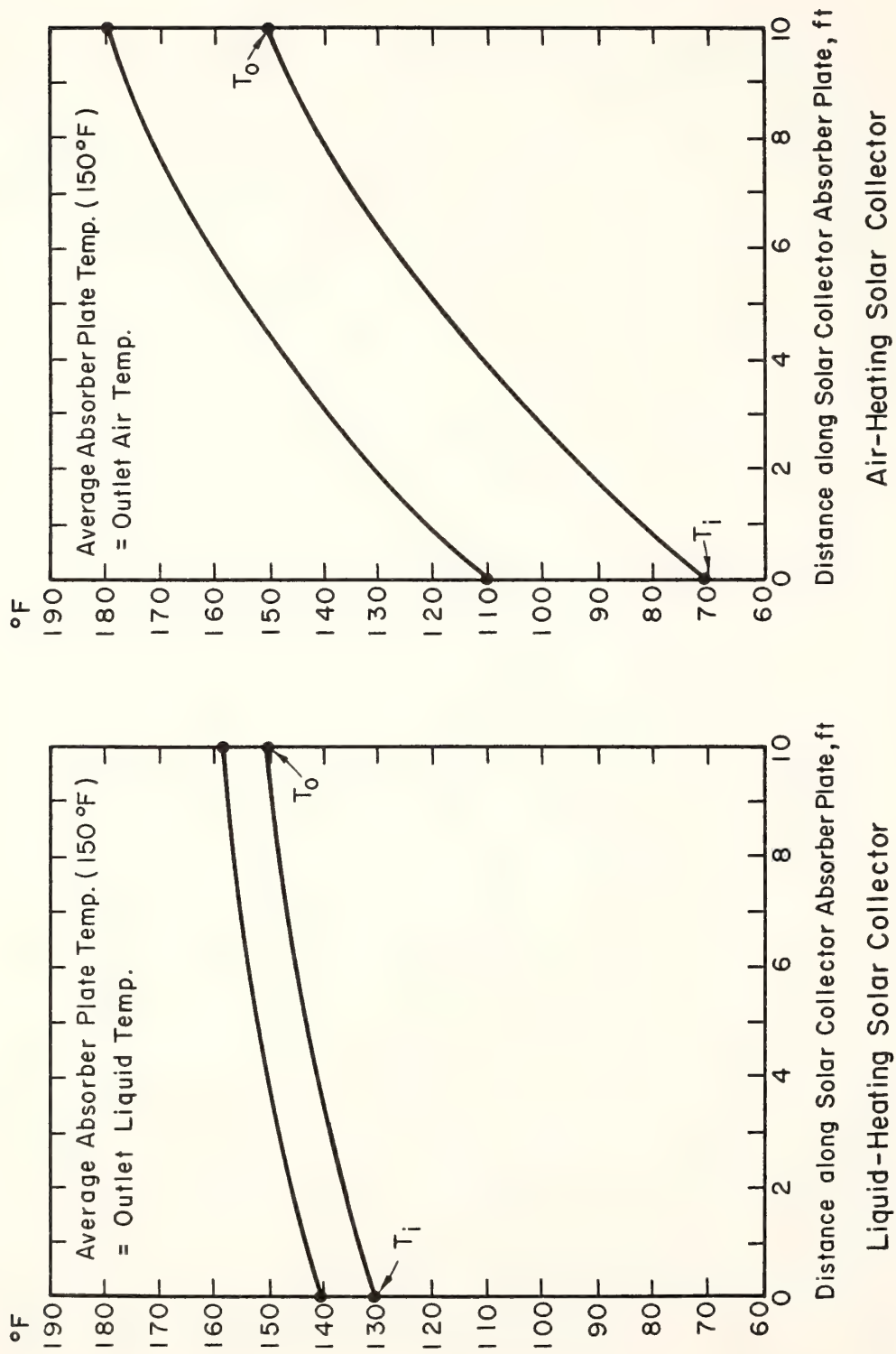


Figure 4-4. Comparison of Typical Temperatures in Liquid- and Air-Heating Solar Collectors

(measured as pounds per hour, for example) for suitable pressure loss conditions. Rather than a moderate 10-degree difference between plate and fluid temperatures, as in the liquid case, the air collector is characterized by a 30- to 50-°F temperature driving force. The much lower heat transfer coefficient from the plate to the air is responsible for this difference. Under the conditions chosen, the average plate temperature would be about 150 degrees, approximately the same as that estimated for the liquid system. Use of Equation (4-2) with an inlet temperature of 70 degrees results in a heat recovery factor, F_R , typically about 0.7 for the air collector. Characteristically, solar air heaters having heat transfer surfaces approximately equal to the solar absorbing area show heat recovery factors substantially below those achieved in liquid collectors. However, as shown below, this difference must not be interpreted as superiority of one over the other when used in suitably designed systems.

COLLECTOR EFFICIENCY

Equation (4-2) may be rewritten as an efficiency of solar collection, η_c , that is, the ratio of useful heat delivery divided by the total solar radiation, by dividing both sides of the equation by I_T and by A_c . Equation (4-3) is the result.

$$\eta_c = \frac{Q_u}{I_T A_c} F_R \tau \alpha - F_R U_L \left[\frac{(T_i - T_a)}{I_T} \right] = \left(\begin{array}{c} \text{collector} \\ \text{efficiency} \end{array} \right) \quad (4-3)$$

For a given collector operating at a constant fluid circulation rate, A_c , F_R , τ , α , and U_L are nearly constant regardless of solar and temperature conditions. Assuming that they are constant, Equation (4-3) represents a straight line on a graph of efficiency versus $T_i - T_a / I_T$ as

shown in Figure 4-5. The characteristics of this line are an intercept (the intersection of the line with the vertical efficiency axis) equal to the numerical value of $F_R \tau \alpha$ and a "slope" of the line, that is, the vertical scale change divided by the horizontal scale change, which is equal to $F_R U_L$. If experimental data on collector heat delivery at various temperatures and solar conditions are plotted on a graph with efficiency as the vertical axis and $T_i - T_a / I_T$ as the horizontal axis, the best straight line through the data points is a complete representation of the collector performance over its entire operating range.

Experimental data are normally taken for collectors with the collector tilted to be perpendicular to the solar beam radiation. Therefore, $F_R \tau \alpha$ from performance curves is labeled $F_R (\tau \alpha)_n$ where $(\tau \alpha)_n$ indicates that the beam radiation is perpendicular to the plane of the collector cover.

The point at which the line intersects the vertical axis corresponds to the fluid inlet temperature being the same as the ambient temperature, and collector efficiency is at its maximum. Where the line intersects the horizontal axis, collection efficiency is zero. This situation corresponds to such a low radiation level or such a high temperature of the fluid supply to the collector that heat losses are equal to solar absorption and no useful heat is delivered from the collector. At this point, the temperature of the absorber plate is equal to T_i , and is the maximum possible at the particular levels of ambient temperature and solar radiation. It is commonly referred to as stagnation temperature of the collector.

In a solar heating system employing a non-freezing liquid and a heat exchanger for transfer of energy to storage, it is convenient

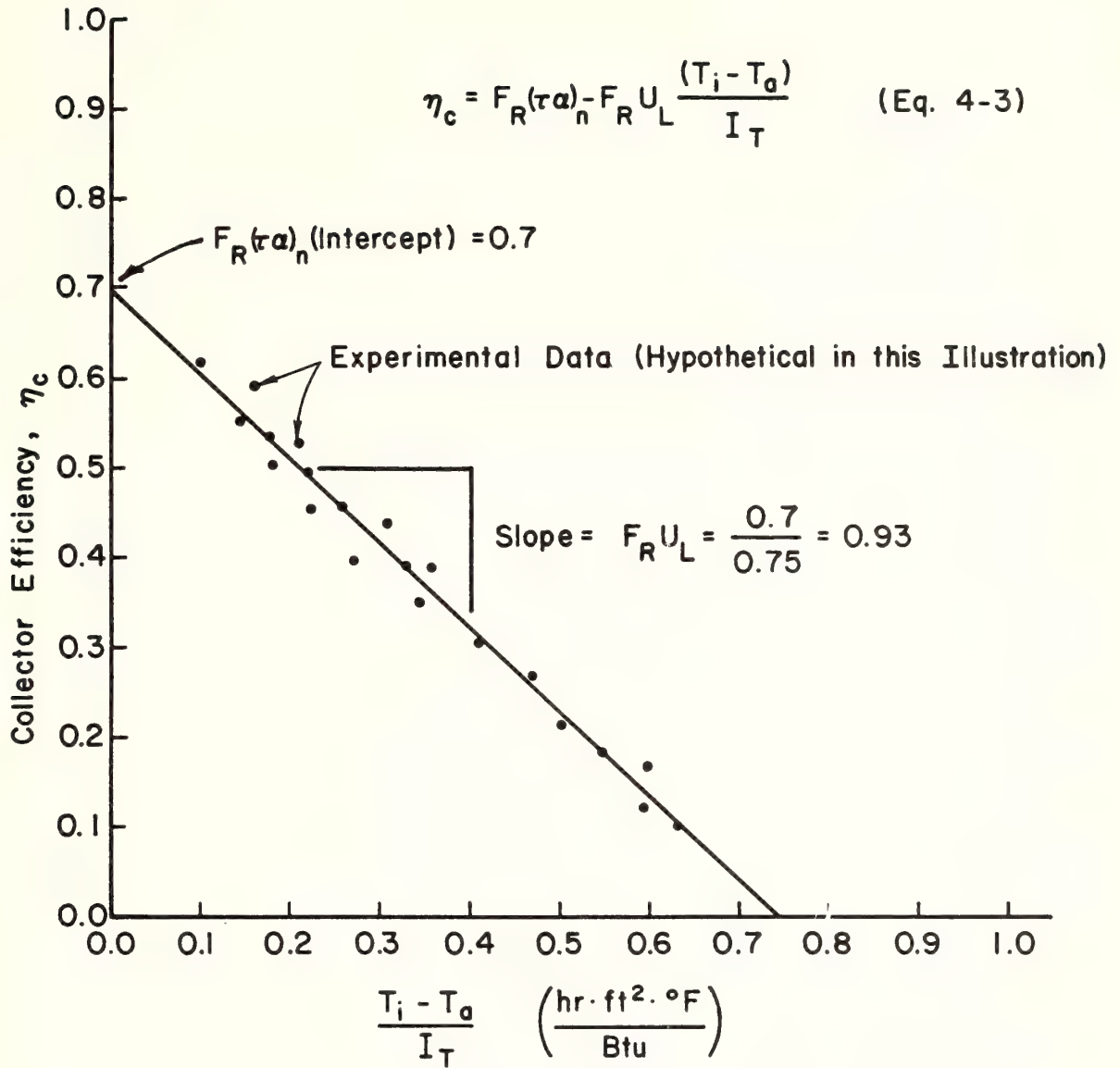


Figure 4-5. Typical Collector Performance Curve

to modify the heat recovery factor, F_R , so that it includes the exchanger and so that storage tank temperature may be substituted for T_i in Equation (4-3). This adjusted factor, F'_R , is less than F_R by the ratio of efficiencies obtainable with and without the exchanger. Its actual value can be determined by use of F_R , the characteristics of the heat exchanger, and the liquid flow rates through the exchanger. The procedure for its determination is outlined in Module 7. For air collectors and for drain-back liquid collectors, no adjustment in F_R is required.

Typical Collector Characteristics

Efficiencies of several types of collectors tested in the NASA Lewis Research Center are shown in Figure 4-6. Collectors 1 and 2 are seen to have the highest efficiencies, but final selection also depends on costs, durability, appearance, and so on. Collector 1 appears to have the best performance of all those compared in Figure 4-6 if normally operated at conditions represented by the left-hand side of the graph. Such conditions are low operating temperatures or high solar radiation. Near the right-hand side of the graph, however, collector 2 is more efficient than collector 1, where high inlet collector temperatures or low solar radiation prevail. It is evident that some collectors are better than others in some temperature and radiation ranges, whereas a reversal can occur at different conditions.

A graph such as Figure 4-6 for comparing a particular collector with others of similar type can be useful in selecting suitable equipment. Collector manufacturers usually provide such data. Of equal value are dependable data on the quantities $F_R(\tau\alpha)_n$ and $F_R U_L$. Knowledge

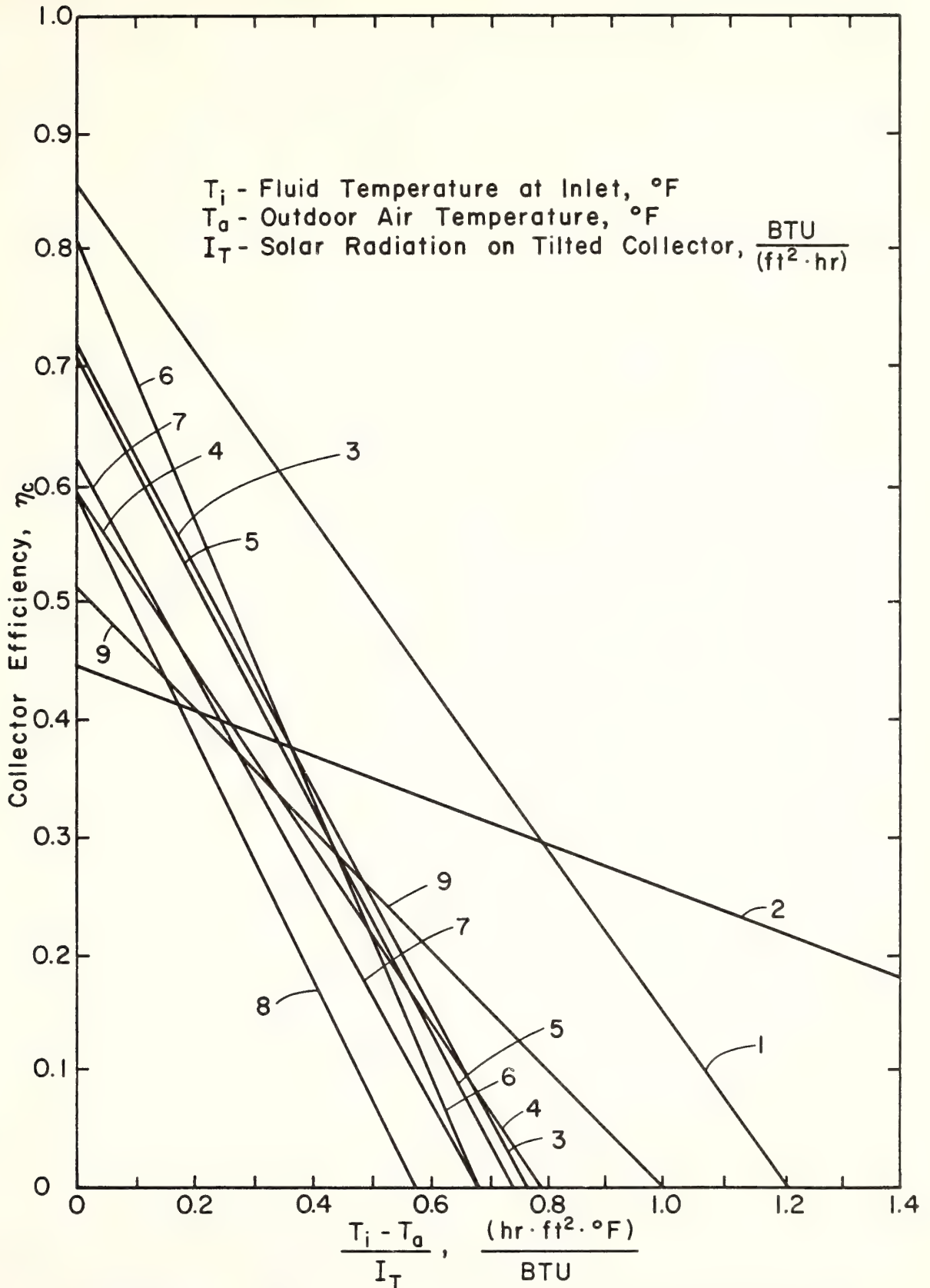


Figure 4-6. Measured Solar Collector Efficiencies of Several Collectors (Number label on graph refers to Reference Number in Table 4-1)

of those two factors is equivalent to having the graphical relationship. Table 4-1 contains this information for the same collectors shown in Figure 4-6 plus other commercially available collectors.

Collector Characteristics Presented with Different Temperature Base

The collector efficiency curves of Figures 4-5 and 4-6 and the characteristic values of $F_R U_L$ and $F_R(\tau\alpha)_n$ listed in Table 4-1 apply specifically when the abscissa (horizontal axis) on the efficiency graph is expressed as the parameter $(T_i - T_a)/I_T$. Collector performance may be correlated with other collector temperatures, and various parameters are used for the abscissa of the collector efficiency curve. Two other parameters that are commonly used are: (1) $[T_i + T_o/2 - T_a]/I_T$ and (2) $(T_o - T_a)/I_T$, where T_o is collector outlet fluid temperature. In the first parameter, $(T_i + T_o)/2$ is used instead of T_i on the premise that an average of the inlet and exit temperatures more nearly represents the average plate temperature as expressed in Equation (4-1). It is also sometimes argued that the fluid temperature measured at the collector exit is more representative of collector performance because it is more indicative of the value or intensity of the heat collected. If the collector temperatures are "well behaved" as illustrated in Figure 4-6, it really makes little difference whether T_i , $(T_i + T_o)/2$, or T_o is used in the parameter to express collector efficiency. However, the heat recovery factor, F_R , will differ for each case.

When either $(T_i + T_o)/2$ or T_o is used in the abscissa of the collector efficiency curve, corrections can be made to F_R , hence to $F_R(\tau\alpha)_n$ and $F_R U_L$. That is, the values of $F_R(\tau\alpha)_n$ and $F_R U_L$ that will

Table 4-1
Collector Performance Parameters*

Ref. No.	Manufacturer	$F_R(\tau\alpha)_n$	$F_R U_L$ (Btu/ft ² h°F)	Type
1	Honeywell (NASA)	0.863	0.715	Liquid
2	Owens-Illinois	0.447**	0.206**	Liquid or Air
3	Miromit	0.724	0.947	Liquid
4	Beasley	0.600	0.759	Liquid
5	Revere	0.716	0.964	Liquid
6	Barber	0.816	1.204	Liquid
7	PPG	0.632	0.930	Liquid
8	Solar Products	0.600	1.057	Liquid
9	Solaron	0.516	0.516	Air
10	Rocky Mountain	0.679**	0.789**	Liquid or Air
11	General Electric	0.639	0.614	Liquid
12	LeRC	0.745	0.820	Liquid
13	InterTechnology	0.650	0.610	Liquid
14	Southwest Std	0.672	0.794	Liquid
15	Sunworks	0.650	0.789	Liquid
16	Trantor	0.700	0.830	Liquid

*From graphical presentation by Johnson, S.M., and Simons, F.F. (1976) "Comparison of Flat-Plate Collector Performance Obtained Under Controlled Conditions in a Solar Simulator". Lewis Research Center, NASA, Cleveland, Ohio. Presented at Annual ISES Meeting, Winnipeg, Manitoba, Canada.

**Numerical values apply to liquid collectors.

later be used in design calculations will be on the uniform basis, as if the parameter $(T_i - T_a)/I_T$ were used to express the collector efficiency curve.

The corrections are as follows:

1. When $(T_i + T_o)/2$ is used,

$$F_R(\tau\alpha)_n [\text{corrected}] = F_R(\tau\alpha)_n [\text{uncorrected}] \times \left[\frac{1}{1 + \frac{(F_R U_L) A_c}{2C_c}} \right]$$

Within the terms in the bracket, $F_R U_L$ is the uncorrected value and C_c is the collector fluid capacitance rate,

$$C_c = \dot{m} c_p \text{ (Btu/hr}\cdot^\circ\text{F)}$$

2. When T_o is used,

$$F_R(\tau\alpha)_n [\text{corrected}] = F_R(\tau\alpha)_n [\text{uncorrected}] \times \left[\frac{1}{1 + \frac{(F_R U_L) A_c}{C_c}} \right]$$

and

$$F_R U_L [\text{corrected}] = F_R U_L [\text{uncorrected}] \times \left[\frac{1}{1 + \frac{(F_R U_L) A_c}{C_c}} \right]$$

where the terms in the brackets are defined under item (1) above.

It should be noted that the mass flow rate and fluid capacitance rate through air collectors can significantly influence the collector performance discussed in the following section. The correction procedures cannot be applied for a flow rate that is different from that used to obtain the experimental data.

Comparison of Liquid and Air Collector Performance

Efficiency relationships for a widely used air collector operating at two different air circulation rates and a liquid collector (collector 10 from Table 4-1) are shown in Figure 4-7. Whereas flow rate

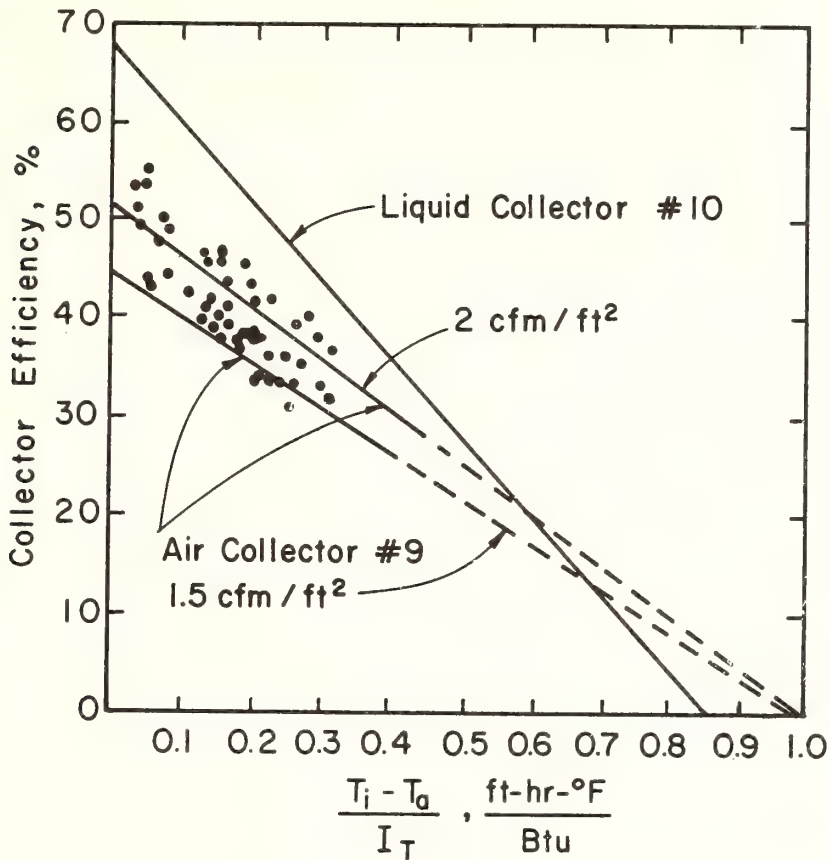


Figure 4-7. Comparison of Liquid and Air Collectors Based on Measured Performance (points shown are for air collector operating at 2 cfm/ft^2)

does not significantly affect the efficiency of a liquid collector, it is evident that air flow rate has a substantial influence on air collector performance. Although even greater efficiencies can be achieved with higher air flow rates, the larger pressure drop and power requirements to circulate air at rates above 2 cfm/ft^2 force a compromise between collector efficiency and power consumption. As seen from Figure

4-7, the liquid collector is more efficient than the air collector (toward the left-hand side) at the same inlet temperature, ambient temperature, and solar radiation level. It is important to recognize, however, that in space heating systems, liquid and air collectors normally operate at very different inlet temperatures so that air collectors usually operate at conditions substantially nearer the left side of the graph than do the liquid type. The net result is comparable operating efficiency of the two collectors when assembled in typical space heating systems. The foregoing comparison leads to the conclusion that whereas similar types of collectors such as flat-plate liquid heaters can be compared by means of a graph such as Figure 4-6, comparisons cannot be drawn in this way between different types. A second conclusion is that since the conditions at which the collector must operate depend on system conditions, particularly storage temperature, comparative evaluation requires attention to the other components in the system and their effect on collector performance.

Table 4-2 contains a step-by-step summary comparison of air and liquid types of collectors. Typical air and water heaters are compared at a high solar radiation level and at a fairly low solar intensity. Characteristic designs and operating conditions have been assumed.

The results of the two calculations are shown in Figure 4-8 in graphical form. It may be noted that at the high solar radiation level, $300 \text{ Btu}/(\text{hr}\cdot\text{ft}^2)$, the two collectors have identical (50 percent) efficiency, and at the lower solar level, $150 \text{ Btu}/(\text{hr}\cdot\text{ft}^2)$, the air collector (operating at the characteristically low return air temperature) has an efficiency substantially greater than the liquid collector.

Table 4-2
Comparison of Typical Solar Heating Systems
Employing Liquid and Air Collectors

Performance Relationship:

Collection Efficiency:

$$\frac{Q_u}{A_c I_T} = F_R \tau \alpha - F_R U_L \left(\frac{T_i - T_a}{I_T} \right)$$

Design Characteristics:

	<u>Liquid</u>	<u>Air</u>
Heat Recovery Factor F_R	0.9	0.7
Heat Loss Coefficient U_L	0.75	0.75
Cover Transmission τ	0.85	0.85
Plate Absorptivity α	0.95	0.95
$F_R(\tau\alpha)_n$	0.73	0.57
$F_R U_L$	0.68	0.53

Operating Conditions:

Atmospheric Temperature T_a , °F	30	30	30	30
Fluid Inlet Temperature T_i , °F	130	130	70	70
Solar Radiation I_T , Btu/(hr·ft ²)	300	150	300	150
Fluid Flow Rate, gpm/ft ² , cfm/ft ²	0.02	0.02	2.0	2.0
$(T_i - T_a)/I_T$	0.333	0.666	0.133	0.266

Calculated Performance:

$F_R U_L (T_i - T_a)/I_T$	0.23	0.46	0.07	0.14
$F_R \tau \alpha - F_R U_L (T_i - T_a)/I_T$	0.50	0.27	0.50	0.43
Collection Efficiency, %	50	27	50	43
Computed Outlet Temperature, °F	145	138	134	125

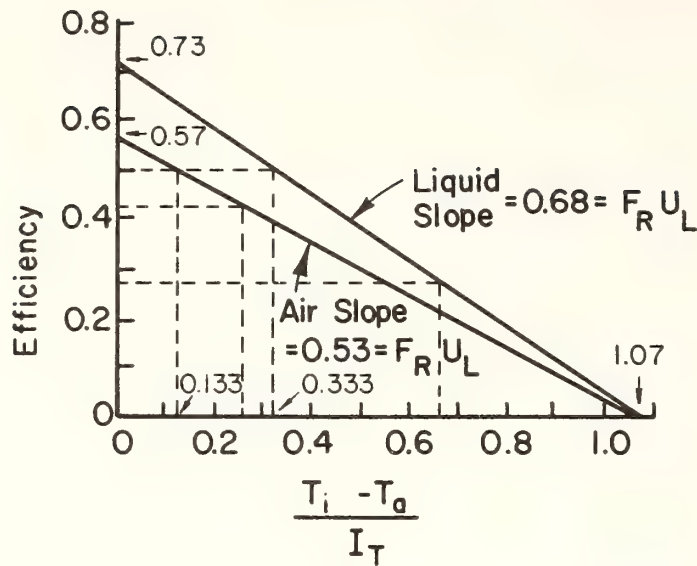


Figure 4-8. Results of Performance Calculations

CORROSION PROTECTION FOR LIQUID COLLECTORS

Corrosion is a major problem that concerns not only collectors but the entire liquid-heating solar system. It is of particular concern in collectors with aluminum absorber plates and systems with aluminum pipes. There are several forms of corrosion that are of concern. These include pitting by oxidation and ion exchange, galvanic corrosion, crevice corrosion, and erosion.

CORROSION BY OXIDATION

Oxidation of metals can be eliminated by removing the dissolved oxygen in the heat transfer liquid and preventing exposure of the liquid to the atmosphere. This can be accomplished in closed systems. While initially there will be free oxygen in the liquid, metal in the closed loop system will react with the oxygen and remove it from the liquid.

The amount of initial oxidation is not of practical concern. In drain-down systems, when fresh air is admitted repeatedly into the system, oxidation is a concern. If drain-down collectors are used, the collector tubing should have high resistance to oxidation. Copper and some types of stainless steel have been found reasonably satisfactory in such usage.

CORROSION BY ION EXCHANGE

Heavy metal ions in the collector fluid (and elsewhere in the system) can cause pitting. Pitting is aggravated by the presence of dissolved chloride ions which may originate from the water supply or from soldering fluxes in the piping. Heavy metal ions may result from corrosion of other parts of the system or may be present naturally in the water supply.

Ion exchange and consequent pitting may be effectively prevented by addition of a small amount of corrosion inhibitor to the transport liquid. Some chemicals that are commonly used as inhibitors are listed in Table 4-3.

It is particularly important that a corrosion inhibitor be added to systems involving antifreeze solutions in water. The most commonly used antifreeze is ethylene glycol. A 50 percent solution by volume of ethylene glycol in water will provide freezing protection down to about -30°F with a boiling point of about 230°F. At high temperatures the glycol can break down to form glycolic acid. This decomposition is accelerated by dissolved oxygen in the system. The resulting acid will reduce the pH of the transport medium and tend to accelerate corrosion. Therefore a solar system that has a glycol antifreeze in the fluid

Table 4-3

Some Inhibitors Suitable for Use in Aqueous Solar
Collector Fluids*

Inhibitor	Function	Suitability	
		Water	Glycol Solution
Sodium Tetraborate $\text{Na}_2 \text{B}_4 \text{O}_7 \cdot 5\text{H}_2\text{O}$	Buffer, protects Fe	?	✓
Sodium Mercaptobenziothiazole (Na MBT) $\text{C}_7\text{H}_4 \text{NH}_2 \text{Na}$	Cu alloy inhibitor	✓	✓
Sodium Metasilicate $\text{Na}_2 \text{SiO}_3 \cdot 9\text{H}_2\text{O}$	General inhibitor for Cu, Fe, Al	✓	✓
Sodium Orthophosphate $\text{Na}_3\text{PO}_4 \cdot 12\text{H}_2\text{O}$	Protects Fe, Al	✓	✓
Sodium Nitrate NaNO_3	Protects Fe, Al, and solder	✓	✓
Sodium Orthoarsenate $\text{Na}_3\text{AsO}_4 \cdot 12\text{H}_2\text{O}$	General inhibitor Most effective on Al	✓	✓
Chromates (Na, K, etc)	General inhibitor for Al, Fe, and Cu	✓	X

*J.M. Popplewell, "Corrosion Prevention in Aluminum Solar Systems",
Paper 170, Corrosion/77, San Francisco, California, March 1977.

Notes:

- (1) Many other inhibitors suitable for solar service are available. This table lists a few of the more common ones only.
- (2) ✓ = Suitable ? = Questionable X = Unsuitable

should be periodically monitored in order to avoid corrosion problems. The pH should be measured regularly and if it deviates more than one pH unit from the original value of the fresh solution, the system should be drained and a new solution should be used. The pH of the original solution can vary with the particular application but is normally recommended to exceed 10.

CORROSION BY GALVANIC ACTION

Galvanic corrosion occurs when two dissimilar metals are joined in an electrolyte. The more noble metal acts as a cathode and is protected by the less noble metal, which acts as an anode and will be consumed. The galvanic relationship between various metals is listed in Table 4-4.

Table 4-4

Galvanic Order of Some Common Metals*
(Electrode Potential is for Sea Water)

Metal	Electrode Potential (Volts) wrt H_2
Magnesium	-1.45
Zinc	-0.80
Aluminum	-0.53
Iron	-0.50
Carbon steel	-0.40
Lead	-0.30
Tin	-0.25
Copper	-0.08
Stainless steel (type 304)	+0.07
Platinum	+0.40

*N.D. Tomashov, Theory of Corrosion and Protection of Metals, MacMillan, 1966.

Although the data were generated for sea water, the metals have the same order of activity in a different electrolyte. It is clear from the data that if copper and aluminum are coupled, the copper will act as a cathode and corrosion will be accelerated on the aluminum.

The best protection against galvanic corrosion is to avoid contact between dissimilar metals. Dielectric couplers (rubber hose) may be used to join two dissimilar metals in the plumbing of a solar system. However, it is important to note that different alloys of the same material may have significantly different electrolytic potentials, and care should be taken to ensure that plumbing fittings are of the same alloy as the pipe.

CORROSION BY CREVICING

Crevice corrosion is similar to pitting corrosion and results in rapid loss of metal inside a crevice. The crevice can be the result of bad fittings, leaky gaskets, scale deposits, blockages, or unusual flow patterns. The crevice represents a restricted area where an occluded cell may develop. The mechanism of crevice corrosion is that oxygen is rapidly depleted inside the cell, thereby creating an anode. If there is abundant supply of oxygen outside the crevice, then the area outside the crevice can serve as a cathode. The small area inside the crevice will become active and rapid corrosion can occur.

Crevice corrosion is perhaps a more difficult type of corrosion to avoid than the previously mentioned corrosion types. The best procedure is to attempt to eliminate crevices through good design, good installation, and filtering to remove any debris that could lead to a blockage.

EROSION OF CONDUITS

Erosion corrosion is the result of mechanical removal of the protective film provided by an inhibitor, and it is most likely to occur when high flow rates and turbulent flow conditions exist. It is aggravated by the entrainment of air and debris.

Control of the flow rate is the best way to avoid metal erosion. A general rule of thumb is to maintain flow rates less than 6 feet per second. It is also recommended that a filter be used to remove debris from the circulating liquid.

FREEZE PROTECTION FOR LIQUID COLLECTORS

ETHYLENE GLYCOL ADDITIVE

Experience indicates that an ethylene glycol concentration of 10 to 20 percent is adequate to prevent pipe and tubing from bursting when exposed to temperatures well below the freezing point of the mixture. If the liquid in the system is static and flow at lower temperatures is not required, glycol concentrations as high as indicated in freezing point tables are not necessary. But pipes leading to a collector must be protected from freezing so that flow is always possible. Otherwise, the liquid in the collector may boil even in midwinter and, if the connecting pipes are frozen, flow cannot occur. Increased pressure in the absorber tubes caused by boiling could burst thin tubes.

Adequate freeze protection for a water collector can be obtained with antifreeze concentrations that are less than those required in an automobile radiator, as the purpose of the antifreeze is to prevent damage to the collector, but not necessarily to prevent the formation

of ice crystals. In Table 4-5, the temperatures and percent ethylene glycol concentration in water (by volume) will result in a slushy

Table 4-5
Concentration of Ethylene Glycol Required for
Freeze Protection

Percent Ethylene Glycol by Volume in Water	Minimum Temperature for Freeze Protection, °F*
0	32
5	26
10	16
15	2
20	-18

*Flow will not be possible below these temperatures

condition which, while very dense, does not result in damage to the absorber tubes. If the corrosion inhibitor additive in the antifreeze is to be effective, the minimum glycol concentration should be about 30 percent. For complete freeze protection in the most severe climates, ethylene glycol concentrations of 50 percent are usually recommended.

In the presence of air, ethylene glycol degrades more readily than without air. A portion of the degradation product results in an acidic solution which promotes corrosion. If simultaneous exposure to oxygen and elevated temperatures of the ethylene glycol solution cannot be avoided, then the temperatures must be moderated. The allowable maximum temperature depends upon the degree of aeration and the desired service life of the solution. A temperature of 250°F may be acceptable when the

only source of air is a vent or vacuum breaker line. Antioxidants are helpful in some applications.

The specific heat, density, and viscosity of aqueous ethylene glycol solutions vary with concentration and temperature. Of particular importance is the reduction of heat capacity and increase in viscosity of the mixture as concentration is increased and temperature is decreased. The effects on the properties of the fluid mixtures are shown in Figures 4-9 through 4-11. In the absence of more detailed information, properties for mixture concentrations different from those shown will necessarily require some interpolative estimations.

ORGANIC FLUIDS FOR FREEZE PROTECTION

Dowtherm J[®] is an organic fluid which has a low freezing point and is also non-corrosive toward all metals or alloys commonly used in solar systems, such as steel, copper, aluminum, and stainless steel alloys. The physical properties of Dowtherm J[®], as well as Therminol 55[®], are listed in Table 4-6. Therminol 55[®] is probably not adequate as a non-freezing fluid in cold climates. Silicone oils, although expensive, are stable, odorless, non-corrosive, and have no freezing or boiling problems.

COLLECTOR ARRAYS

The previous sections concerned individual collector modules or panels, and, in general, several modules are required in a solar system to provide the energy to meet the heating needs. Collector modules may

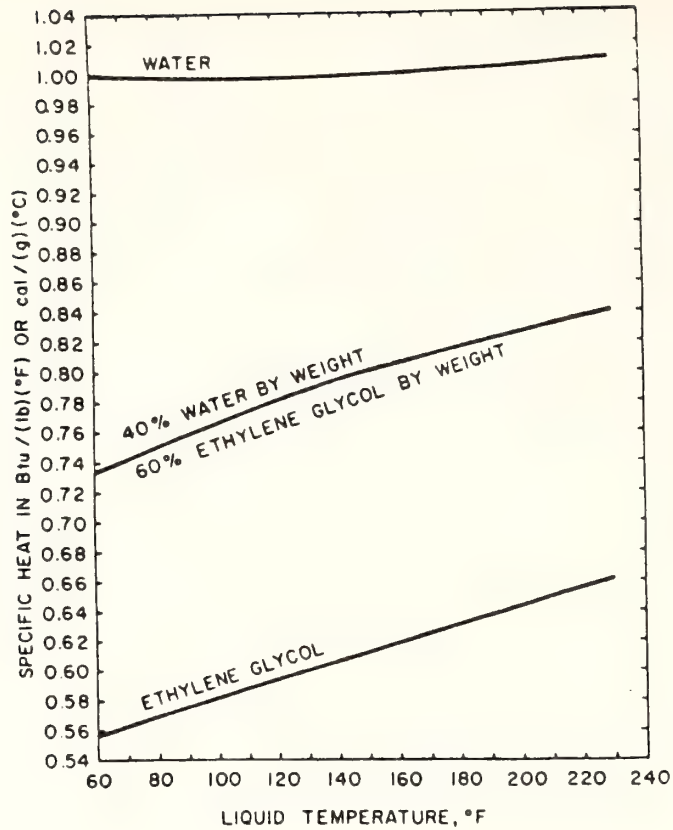


Figure 4-9. Specific Heat of Aqueous Ethylene Glycol Solutions

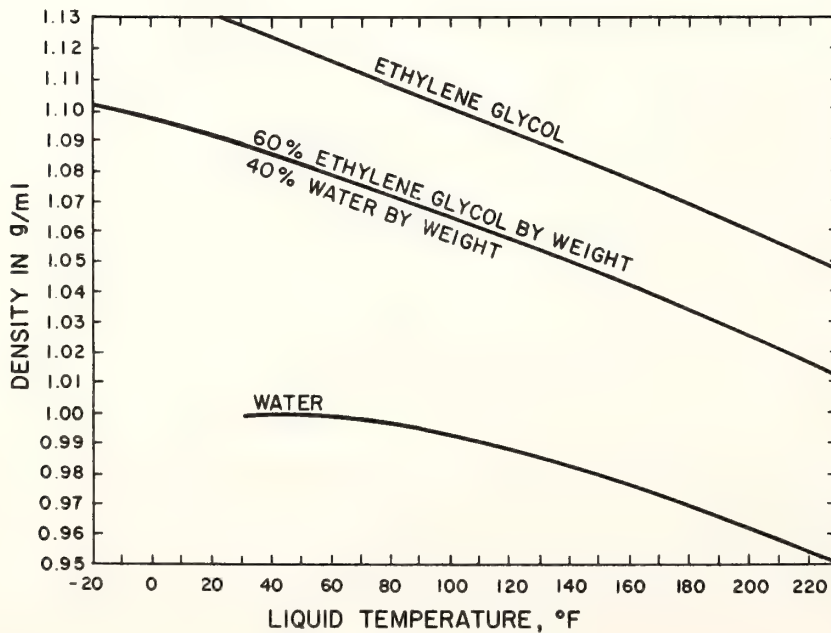


Figure 4-10. Density of Aqueous Ethylene Glycol Solutions

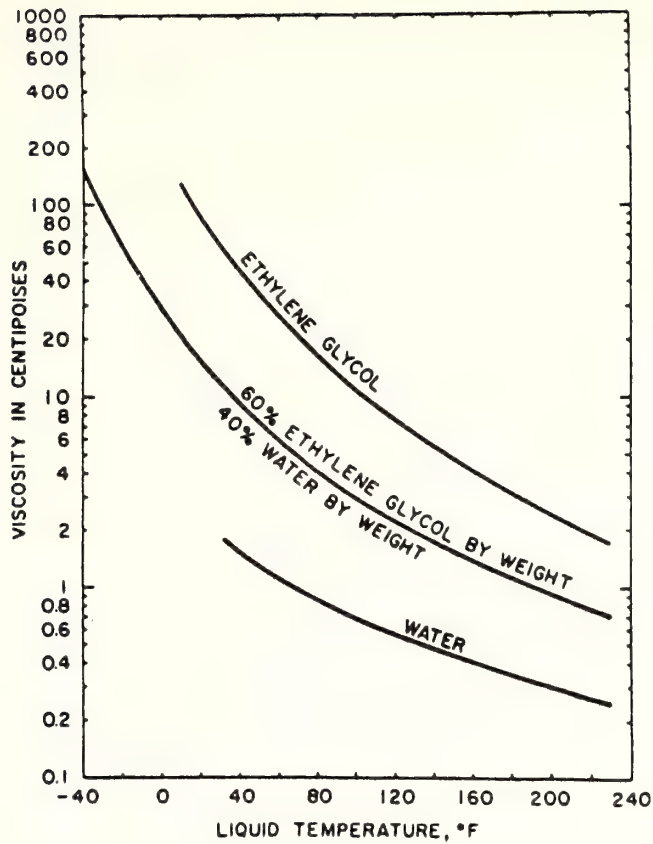


Figure 4-11. Viscosity of Aqueous Ethylene Glycol Solutions

Table 4-6

Some Physical Properties of Dowtherm J[®] and Therminol 55[®]

Property	Dowtherm J [®]	Therminol 55 [®]
Operating temperature range, °F	-100 to 575	-5 to 600
Pour point, °F	---	-40
Boiling point, °F	358	635
Flash point, °F	145	355
Fire point, °F	155	410
Auto ignition temperature, °F	806	675

be assembled in parallel and in series to form a complete array. An important consideration is a design by which nearly equal flow through each collector can be obtained. Liquid-heating solar collectors may be arranged as shown in Figure 4-12 with headers at the top and bottom of the array. Satisfactory flow distribution is realized if the headers are large enough for the head loss (pressure drop) from the bottom to the top of each collector (column) to be above 95 percent of the total head loss from A to B.

An arrangement for an array of air-heating collectors is shown in Figure 4-13. The main manifold ducts are sized in accordance with the volumetric air flow rates and, in the scheme shown, the manifolds within the collectors are used as "headers". The specific arrangement depends upon the design of the collector.

Collector arrays, for both liquid- and air-heating systems, should be leak tested during or immediately after assembly. While leaks in liquid systems are easy to detect, leaks in air systems are not as readily located. Care in the installation of collectors, pipes, and ducts, is essential. The cost of labor for careful assembly is a small fraction of the cost of labor for disassembly and making repairs.

Joints in piping, particularly from the headers to the collector modules, may be made with flexible hoses. Because neoprene or rubber hoses require periodic replacement, the design should make replacement convenient. Other piping connections and valve locations should be given similar consideration. Connections from the headers to the absorber plates of liquid collectors cannot usually be rigid couplings because of thermal expansion and contraction of the plates and headers. A number of manufacturers recommend all-metal connections (for

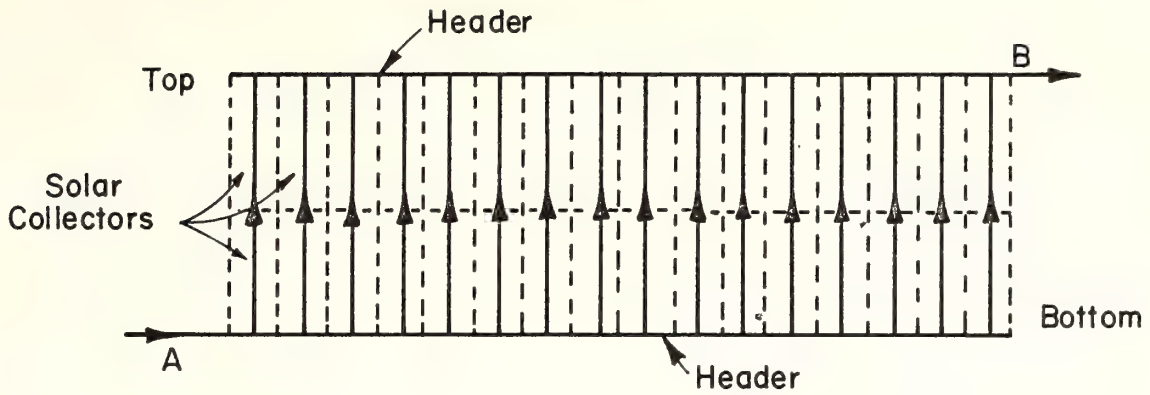


Figure 4-12. Definition Sketch for Fluid Flow Distribution Through a Solar Collector Array

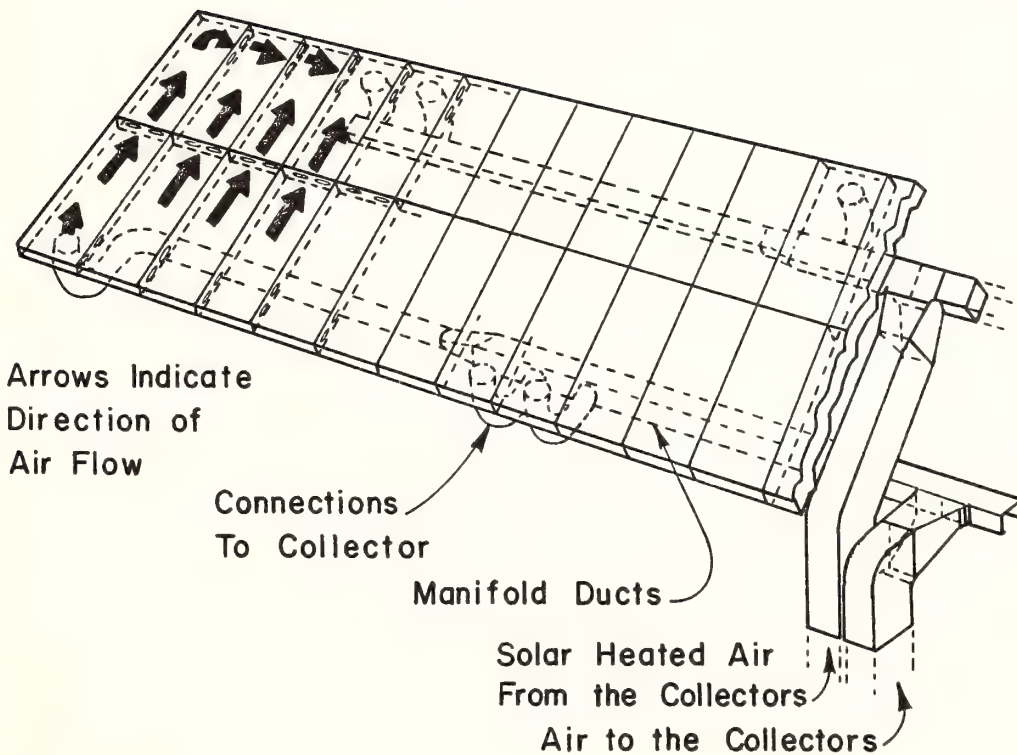


Figure 4-13. Typical Arrangement of Internally Manifolded Collector Modules in an Array

durability) to the solar panels, with sufficient flexibility in piping and manifolding to permit expansion and contraction. Air collector modules may be connected by flexible ducting or by gasketed ports.

REFERENCES

1. Dickinson, W.S., Neifert, R.D., Löf, G.O.G., and Winn, C.B., (1975). "Performance Handbook for Solar Heating Systems". Presented at the 1975 International Solar Energy Society Congress, University of California at Los Angeles, California.
2. Oonk, R.L., Löf, G.O.G., and Shaw, L.E., (1976). "A Method of Comparing Flat-Plate Air and Liquid Solar Collectors for Use in Space Heating Applications". Presented at the 1976 International Solar Energy Conference, Winnipeg, Manitoba, Canada.
3. Klein, S.A., Beckman, W.A., and Duffie, J.A., (1975). "A Design Procedure for Solar Heating Systems". Presented at the 1975 International Solar Energy Society Congress, University of California at Los Angeles, California.
4. Johnson, S.M. and Simon, F.F., (1976). "Comparison of Flat-Plate Collector Performance Obtained Under Controlled Conditions on a Solar Simulator". Presented at the 1976 International Solar Energy Society Annual Meeting of the American Section, Lewis Research Center, Cleveland, Ohio.
5. Palerman, L.D., (1959). "The Physical Properties and Behavior of Ethylene and Propylene Glycol and Their Water Mixtures". Presented at the Annual Meeting of the American Society of Heating and Air Conditioning Engineers, Incorporated, Philadelphia, January 26-29.
6. Private communication with D.J. Fulginiti, Antifreeze Marketing Consumer Products Division, Fabrics and Finishes Department, E.I. DuPont De Nemours and Company, Incorporated, Wilmington, Delaware, 19898.
7. Ward, J.C. and Löf, G.O.G., (1976). "Long-Term (18 Years) Performance of a Residential Solar Heating System". Solar Energy, Vol. 18, No. 4, pp. 301-308.
8. Duffie, J.A. and Beckman, W.A., (1974). Solar Energy Thermal Processes, John Wiley and Sons, New York.
9. Popplewell, J.M. (1977). "Corrosion Prevention in Aluminum Solar Systems", Paper 170, Corrosion/77, San Francisco, California, March 1977.
10. Tomashov, N.D., (1966). Theory of Corrosion and Protection of Metals, MacMillan and Company.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 5

HEATING AND COOLING LOAD ANALYSES

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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GLOSSARY OF TERMS

Cooling Load	Rate of heat removal from a building to maintain constant indoor temperature.
Conductance	Expresses the ease or difficulty of materials to conduct heat from the high temperature side to the low temperature side.
Design Equivalent Temperature Differential	Temperature differential between outdoor and indoor temperature adjusted for effects of surface absorptance of solar energy and radiation.
Heat Gains	Rate of heat flow into a building per unit time, usually one hour.
Heat Loss	Rate of heat flow out of a building per unit time, usually one hour.
Heat Transmission Losses	Heat loss.
Heating Load	Rate of heat supply to a building to maintain constant indoor temperature.
Infiltration Gain	Infiltration of hot air into a building which must be cooled to the comfort level of the building air.
Infiltration Loss	Infiltration of cold air into a building which must be heated to the comfort level of the building air.
R value	Thermal resistance of materials to flow of heat.
U value	Heat transfer coefficient for material.

LIST OF SYMBOLS

A	Area of heat transfer surface, ft^2
f_i	Air film conductance for still air on inside surface of the building, $\text{Btu}/(\text{hr})(\text{ft}^2)(^\circ\text{F})$
f_o	Air film conductance for moving air on outside surface of the building, $\text{Btu}/(\text{hr})(\text{ft}^2)(^\circ\text{F})$
Q	Heat flow rate, Btu/hr
R	Thermal resistance of building materials, $(\text{hr})(\text{ft}^2)(^\circ\text{F})/\text{Btu}$
R_i	Thermal resistance of inside air film, $(\text{hr})(\text{ft}^2)(^\circ\text{F})/\text{Btu}$
R_o	Thermal resistance of outside air film, $(\text{hr})(\text{ft}^2)(^\circ\text{F})/\text{Btu}$
R_t	Total thermal resistance for composite building components, $(\text{hr})(\text{ft}^2)(^\circ\text{F})/\text{Btu}$
T_i	Design inside temperature, $^\circ\text{F}$
T_G	Design garage temperature, $^\circ\text{F}$
T_o	Design outdoor temperature, $^\circ\text{F}$
U	Heat transfer coefficient, $\text{Btu}/(\text{hr})(\text{ft}^2)(^\circ\text{F})$
U_o	Overall heat transfer coefficient, $\text{Btu}/(\text{hr})(\text{ft}^2)(^\circ\text{F})$
V	Volume change of air in the rooms per hour, ft^3/hr
W_i	Humidity ratio of indoor air, dimensionless
W_o	Humidity ratio of outdoor air, dimensionless

OBJECTIVE

The objective of this module is to present methods for estimating heat losses and gains in buildings. From the information contained in this module the trainee should be able to:

1. Determine heat transmission coefficients.
2. Compute heat transmission losses.
3. Compute the heating load of a building.
4. Compute the cooling load of a building.

INTRODUCTION

Proper sizing of space heating and air-conditioning equipment requires knowledge of heating and cooling loads in a building. This applies to solar systems as well as conventional equipment. Oversizing of conventional space-conditioning equipment which has been prevalent in the past, especially for residential buildings, leads to inefficient operation. On the other hand, undersizing is not acceptable from a standpoint of comfort. Oversizing of solar systems is not advisable because of high initial cost, but undersizing may not provide significant savings of conventional fuels. It is therefore important to make accurate calculations of heating and cooling loads for proper sizing of solar as well as conventional equipment.

HEAT LOSSES

Heat transmission losses, or more simply heat losses, from buildings occur in two significant ways: (1) transmission losses through walls, floor, ceiling, glass and other surfaces and (2) infiltration losses through cracks and crevices in the walls and through open doors and windows.

HEAT TRANSMISSION THROUGH BUILDING ENVELOPE

The rate of heat flow, Q , into or out of a building enclosure depends upon the surface area, A , an overall heat transfer coefficient, U , and the air temperature difference between the inside, T_i , and outside, T_o . Expressed in equation form:

$$Q = UA(T_i - T_o) \quad (\text{Btu/hr}) \quad (5-1)$$

The overall heat transfer coefficient, often called the U factor, is the reciprocal of the total thermal resistance, R_T , and is expressed as

$$U = \frac{1}{R_T} \quad [\text{Btu}/(\text{hr} \cdot \text{ft}^2 \cdot ^\circ\text{F})] \quad (5-2)$$

and

$$R_T = R_1 + R_2 + R_3 + R_4 + \dots + R_n \quad (5-3)$$

where the R Factors R_1 , R_2 , etc., are individual resistances of the wall components.

Heat is transferred from inside air to the inside wall surface through a thin film of air adjacent to the wall surface. This air film has a resistance, R_i , which is the reciprocal of the film conductance, f_i , and may be determined by

$$R_i = \frac{1}{f_i} . \quad (5-4)$$

Similarly, there is a thin air film at the outside surface, with conductance f_o , which provides some resistance to heat flow. The resistance of the outside air film, R_o , is

$$R_o = \frac{1}{f_o} . \quad (5-5)$$

Surface conductances and resistances for air films for interior and exterior surfaces, in winter and summer, are tabulated in Table A5-1 of the Appendix to this module. The winter values are based on wind velocity of 15 mph and summer values are based on wind velocity of 7 mph.

Dead air spaces between walls offer thermal resistance. The resistance values are tabulated in Table A5-2 for 3/4-inch and 4-inch spaces for winter and summer conditions. For spaces between 3/4 inch and 4 inches, values may be interpolated.

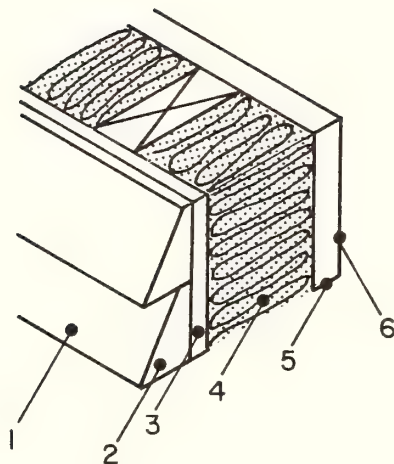
Thermal resistances of common building materials are tabulated in Table A5-3. U factors for windows and patio doors are listed in Table A5-4, and U factors for solid doors with and without storm doors are in Table A5-5. The values in these tables correspond with more complete tables listed in Chapter 20, ASHRAE Handbook of Fundamentals (1972).

TRANSMISSION COEFFICIENTS

The procedure for determining the overall heat transmission coefficients, U, for typical wall, roof, ceiling and floor construction is presented in this section. The values of R for materials and components are found in Tables A5-1 through A5-5. U factors for composite construction are determined in the following examples and U factors for other types of construction may be calculated by following these examples.

Example 5-1 - Frame Wall (2 x 4 studs)

<u>ITEM</u>	<u>R</u>
1. Outside film (15 mph wind, winter)	0.17
2. Siding, wood ($\frac{1}{2}$ x 8 lapped)	0.81
3. Sheathing ($\frac{1}{2}$ inch regular)	1.32
4. Insulation batt (3-3 $\frac{1}{2}$ inch)	11.00
5. Gypsum wall board ($\frac{1}{2}$ inch)	0.45
6. Inside surface (winter)	<u>0.68</u>
Total Resistance, R_T	14.43
$U = 1/R_T$	0.07



The calculated U factor applies to the area between 2 x 4 studs. Because the resistance to heat flow through the 2 x 4 stud is different from the insulation, a correction is sometimes considered. However, the correction is usually small, amounting to less than the accuracy of the R values, and therefore unnecessary.

Example 5-2 - Frame Wall (2 x 6 studs)

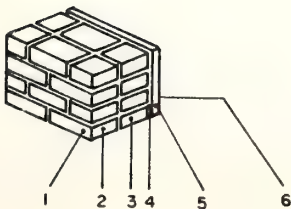
From Example 5-1

R_T	14.43
Replace 3½-inch insulation, subtract	<u>11.00</u>
Add 5½-inch insulation	<u>3.43</u>
	<u>19.00</u>
New R_T	22.43
$U = 1/R_T$	0.04
Difference in U from Example 1	0.03
Percent Difference from 2 x 4 wall	43 percent

There is 43-percent reduction in heat loss for a 2 x 6 wall as compared with a 2 x 4 wall with correspondingly thicker insulation in the 2 x 6 wall.

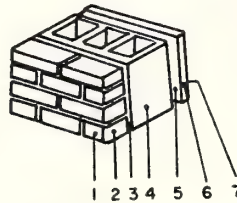
Example 5-3 - Solid Masonry Wall

<u>ITEM</u>	<u>R</u>
1. Outside film (15 mph wind, winter)	0.17
2. Face brick (4 inch)	0.44
3. Common brick (4 inch)	0.80
4. Air space (¾ inch)	1.28
5. Gypsum board (½ inch)	0.45
6. Inside surface	<u>0.68</u>
Total Resistance, R_T	3.82
$U = 1/R_T$	0.26

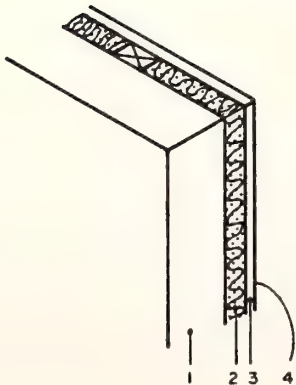


Example 5-4 - Masonry Walls

<u>ITEM</u>	<u>R</u>
1. Outside surface (15 mph)	0.17
2. Face brick (4 inch)	0.44
3. Cement mortar ($\frac{1}{2}$ inch)	0.10
4. Cinder block (8 inch)	1.72
5. Air space ($\frac{3}{4}$ inch)	1.28
6. Gypsum board ($\frac{1}{2}$ inch)	0.45
7. Inside surface	<u>0.68</u>
Total Resistance, R_T	4.84
$U = 1/R_T$	0.21

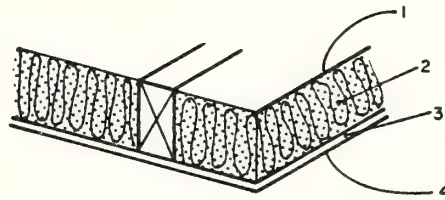
Example 5-5 - Basement Wall

<u>ITEM</u>	<u>R</u>
1. Concrete wall (8 inch)	0.64
2. Insulation batt (2 inch)	7.00
3. Gypsum board ($\frac{1}{2}$ inch)	0.45
4. Inside surface	<u>0.68</u>
Total Resistance, R_T	8.77
$U = 1/R_T$	0.11



Example 5-6 - Insulated Ceiling, 6 inches

<u>ITEM</u>	<u>R</u>
1. Inside surface	0.68
2. Insulation batt (6 inch)	19.00
3. Gypsum board ($\frac{1}{2}$ inch)	0.45
4. Inside surface	<u>0.68</u>
Total Resistance, R_T	20.81
$U = 1/R_T$	0.05

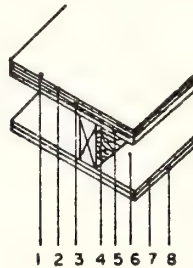
Example 5-7 - Insulated Ceiling, 9 inches

<u>ITEM</u>	<u>R</u>
1. Inside surface	0.61
2. Insulation (9 inch)	24.00
3. Gypsum board ($\frac{1}{2}$ inch)	0.45
4. Inside surface	<u>0.61</u>
Total Resistance, R_T	25.67
$U = 1/R_T$	0.04

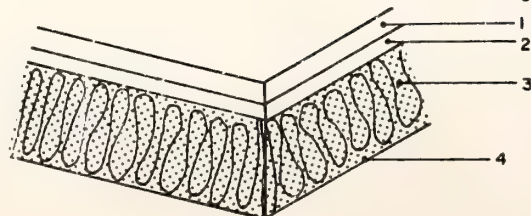
Percent decrease of U with 9-inch insulation over
6-inch insulation, 20 percent

Example 5-8 - Floor

<u>ITEM</u>	<u>R</u>
1. Top surface	0.61
2. Linoleum or tile	0.05
3. Felt	0.06
4. Plywood (5/8 inch)	0.78
5. Wood subfloor (3/4 inch)	0.94
6. Air space	0.85
7. Acoustic ceiling tile (3/4 inch)	1.89
8. Surface	<u>0.61</u>
Total Resistance, R_T	5.79
$U = 1/R_T$	0.17

Example 5-9 - Floor

<u>ITEM</u>	<u>R</u>
1. Carpet and fibrous pad	2.08
2. Plywood (3/4 inch)	0.93
3. Insulation (9 inch)	24.00
4. Surface (still air)	<u>0.61</u>
Total Resistance, R_T	27.62
$U = 1/R_T$	0.04

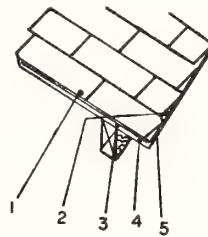


Example 5-10 - Basement Walls and Floor

A heat transfer coefficient, U , for basement walls and floors of $0.10 \text{ Btu}/(\text{hr} \cdot \text{ft}^2 \cdot ^\circ\text{F})$ is generally conservative because (dry) earth adds to overall thermal resistance. A ground temperature equal to the ground water temperature (generally 45°F to 55°F) is appropriate for use as the outside temperature in Equation (5-1).

Example 5-11 - Pitched Roofs (Heat Flow Up)

<u>ITEM</u>	<u>R</u>
1. Outside surface (15 mph)	0.17
2. Asphalt shingle roofing	0.44
3. Building paper	0.06
4. Plywood deck (5/8 inch)	0.78
5. Inside surface	<u>0.61</u>
Total Resistance, R_T	2.06
$U = 1/R_T$	0.49

HEAT LOSS BY INFILTRATION

There are two methods for estimating infiltration losses, the "crack" method and the "air change" method. Of the two methods, the air change method is simpler and easier to use and is the one discussed in this module. Details of the crack method are explained in the

ASHRAE Handbook of Fundamentals (1972). In either method, the objective is to determine the amount of heat required to raise the temperature of cold air which enters a building through cracks, open windows, and doors, to room temperature.

The volume of cold air expected to enter a building through cracks during a one-hour period depends on such factors as wind direction and speed, pressure differences inside and outside the building, whether there are storm windows and doors or air locks on outdoor entrances. The entering volume of cold air may be expressed in terms of the volume of the room or the building interior and the number of air changes per hour. The average air changes for rooms with various fenestrations listed in Table A5-6 are in accordance with Chapter 19, ASHRAE Handbook of Fundamentals (1972).

From the air change rate the contribution to the heating load by infiltration is calculated from

$$Q = 0.018 V (T_i - T_o) \quad (5-6)$$

where V is the volume change per hour (ft^3/hr) and Q is the increase in heating load, Btu/hr.

When moisture is added to air to maintain winter comfort conditions, heat will be required to evaporate water which adds to the building heating load. The added heating load is:

$$Q = 79.5 V (W_i - W_o) \quad (5-7)$$

where V is the infiltration rate (ft^3/hr), W_i is humidity ratio of indoor air, and W_o is humidity ratio of outdoor air.

DESIGN TEMPERATURES

The "design" indoor temperature, T_i , in Equation (5-1) is somewhat arbitrary, with 70°F commonly used for heating load calculations. When the procedure for heat load calculations was developed during the 1940's and early 50's to size natural gas furnaces, the design indoor temperature was 75°F. With modern building construction practices and materials a lower indoor temperature may be used. The lower indoor temperature leads to smaller design heat loss rates and smaller furnaces.

The design outdoor temperature, T_o , is based on a statistical analysis of the hourly temperature readings in December, January, and February. For residential buildings a temperature is selected such that 99 percent of the time during those months, the air temperature is higher than that value. For non-residential buildings a 97.5 percent temperature is recommended. Design temperatures for residential building calculations are listed for various cities in Table A5-10 in the Appendix.

Temperatures of Unheated Spaces

Attic Temperature - The attic temperature is determined from a steady-state balance of heat flow into and out of the attic. Heat flow into the attic is from the ceiling; heat flow out is through the roof surfaces and end walls. The general formula for determining attic temperature is:

$$T_{at} = \frac{A_c U_c T_c + T_o (A_r U_r + A_w U_w)}{A_c U_c + A_r U_r + A_w U_w} \quad (5-8)$$

where

T_{at}	is attic temperature, °F
T_c	is room temperature, °F
T_o	is outside temperature, °F
A_c	is ceiling area, ft^2
A_r	is roof area, ft^2
A_w	is roof wall area, ft^2
U_c	is ceiling U factor, $Btu/(hr)(ft^2)(°F)$
U_r	is roof U factor, $Btu/(hr)(ft^2)(°F)$
U_w	is wall U factor, $Btu/(hr)(ft^2)(°F)$

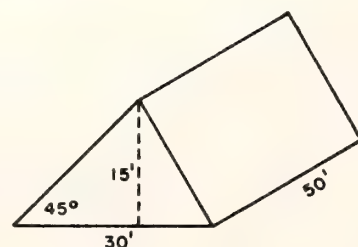
Example 5-12 - Attic Temperature for a Wood Shingled Roof

Calculate attic temperature for a wood shingled roof with the given dimensions. T_o is $-9°F$, T_c is $68°F$. See Example 5-6 for ceiling U factor, $U_c = 0.05$. See Example 5-11 for roof U factor, $U_r = 0.49$. For Example 5-1 a wall with no insulation has a U factor of:

R_T from Example 5-1	14.43
Subtract insulation	-11.00
Subtract gypsum board	<u>- 0.45</u>
Total Resistance, R_T	2.98
$U_w = 1/R_T$	0.34

Calculated Area:

$$\begin{aligned}
 A_c &= 30 \times 50 = 1500 \text{ ft}^2 \\
 A_r &= \sqrt{2} \times 15 \times 50 \times 2 = 2120 \text{ ft}^2 \\
 A_w &= 30 \times 15 \times \frac{1}{2} \times 2 = 450
 \end{aligned}$$



$$T_{at} = \frac{(1500)(0.05)(68) + (-9)[(2120)(0.49) + (450)(0.34)]}{(1500)(0.5) + (2120)(0.49) + (450)(0.34)}$$

$$T_{at} = \frac{5100 - 10,726}{75 + 1039 + 153} = -2.9^{\circ}\text{F}.$$

When ventilation is provided, at 0.5 cfm per square foot of ceiling, the attic temperature must be reduced from those calculated in Example 5-12. Thus, the attic temperature approaches outdoor temperature. Attic temperature may be assumed to be the outdoor temperature with well-insulated ceilings without significant error in heat loss calculation.

Unheated Garage - With similar detailed calculations it can be shown that the temperature in an unheated garage is nearly equal to the mean of the indoor and outdoor temperatures. Since the garage temperature is subject to large changes, a simple calculation to estimate the garage temperature is satisfactory, and T_G is used in Equation 5-1 in place of T_o . An unheated garage temperature may be estimated from

$$T_G = \frac{T_o + T_i}{2} . \quad (5-9)$$

HEATING LOAD

An example heat loss calculation is presented for a house in Fort Collins, Colorado. The building and plans are shown in Figure 5-1, and the description of materials is given in Table 5-1. The windows in all bedrooms are 3' x 4', double hung, single pane, wood sash with storm windows having 3-inch air space. The window in the bathroom is 2' x 2',

double hung, single pane, wood sash with storm window. The window in the living room is 4' x 8', wood sash, double glass with $\frac{1}{2}$ -inch air space. The window in the kitchen is 2.5' x 4' double hung, single pane, wood sash with storm window. The window in the breakfast nook is 3' x 4' double glass, wood sash with $\frac{1}{2}$ -inch air space. The 6' x 6' sliding patio door in the family room is double-glass wood frame with $\frac{1}{2}$ -inch air space. The basement windows are $1\frac{1}{2}$ ' x $1\frac{1}{2}$ ' and will be ignored in this calculation. Bathrooms and kitchen are ventilated.

A worksheet is used to facilitate heat load calculations. The design winter temperature for Fort Collins, Colorado, is not listed in Table A5-10, but is taken to be -9°F . The design indoor temperature is chosen to be 68°F . The total heat loss rate from the building using the design temperatures is 53,215 Btu per hour (see page 2 of worksheet).

Heating loads for each month of the year may be calculated from the heat loss rate and information on the average number of degree days in the month. A degree-day is the difference between 65°F and the average temperature during a 24-hour period, where average temperature is the mean of the high and low temperature. Thus if the average temperature for a 24-hr period is 64°F , there would be one degree-day for that day. The sum of the degree days for each day of the month results in the total degree days for the month, and the sum of the degree days in each month results in the annual degree-days. Monthly and annual degree days for a number of cities in the United States are listed in Table A5-10. Maps of heating degree days are also provided in Figures A5-1 through A5-12 and may be used for locations that are not tabulated in Table A5-10.

To calculate monthly and annual heating loads, calculate the unit heating load for the building, Q_{DD} , from Equation (5-10)

$$Q_{DD} = \frac{\text{Heat Loss Rate}}{\text{Design Temperature Diff.}} \times 24(\text{Btu/DD}) \quad (5-10)$$

For the example building, the unit heating load is

$$Q_{DD} = \frac{53,215 \times 24}{68 - (-9)} = 16590 \text{ Btu/DD}$$

and using the annual heating degree days for Denver, from Table A5-10, the annual heating load for the building, L , is

$$L = 16590 \times 6283 = 104.2 \text{ MMBtu}$$

where MMBtu is an abbreviation for million Btu.

Using the calculations performed above, the furnace for the building should have a heat output capability of about 55,000 Btuh. It will be noted that the contractor has specified a hot water heating system with a 120-150 Btuh capability which is 2.5 to 3 times the required capacity, and leads generally to inefficient water boiler operation because of excessive on-off cycling for short intervals of operation at lowered efficiency.

HEAT GAINS

TRANSMISSION

Heating of air inside a building takes place by radiation and conduction from building surfaces and by infiltration of warm air into conditioned space. The detailed procedure for calculating heat gains into a building is quite complex, taking into account the thermal and optical properties of the building materials, time of day, day of the year, solar radiation intensity, etc. The procedure described in this

module is based on a simplified method using a design equivalent temperature difference (DETD).

Heat gain is then computed by:

$$Q = UA(DETD) \quad (5-11)$$

where

Q is rate of heat gain, Btu/hr

A is area of surface, ft²

U is heat transmission coefficient, Btu/(hr·ft²·°F)

DETD is design equivalent temperature difference.

The DETD's for three design outdoor temperatures are listed in Table A5-7. U factors for typical construction may be computed in the manner shown in Examples 5-1 through 5-11. Heat gain through windows depends upon exposure to the sun and will differ for different window orientations as listed in Table A5-8. No credit is given for shade line below an overhang in the table. When a permanent overhang is provided, the shaded window may be treated as a north-facing window. Average shade lines below an overhang for various latitudes and window orientations are given in Table A5-9. The overhang width multiplied by the shade factor determines the average effective shadow lines below the level of the overhang. Data are for August 1, averaged over 5 hours.

INFILTRATION

Infiltration in the summer is less than in winter because the temperature difference and wind velocity are usually less. Air changes per hour for the summer are listed in Table A5-6. Sensible heat gain is

determined by Equation (5-7) and latent heat gain by Equation (5-8). Residential cooling loads are almost always based only on sensible heat gains.

OCCUPANCY

Heat gain from human occupancy in a residence is usually assumed to be about 300 to 400 Btu per hour per person. For normally equipped kitchens, heat gains from electrical appliances may amount to 1200 Btuh although larger values may be applicable for homes with many appliances.

SOLAR EQUIPMENT

Heat gains (losses) from solar equipment in a residence, such as storage tanks, motors, heated pipes and ducts, will add to the cooling load. The heat gain could be significant from water storage tanks if the insulation is inadequate and the equipment room is not vented. A heat gain equivalent to the kitchen load, 1200 Btuh, may be assumed if the solar storage tank is used during the summer.

LATENT HEAT

A latent heat load of 30 percent of the sensible heat load is generally applicable for residential buildings.

COOLING LOAD

The distinction between maximum heat gain rate and cooling load is important in selecting the size of air-conditioning equipment. There are relatively few days each season with high heat gains, and a partial load condition exists for many hours during a season. Thus, cooling

equipment sized on the basis of maximum heat gain rate would be oversized, and would not perform efficiently because of intermittent cycling. Equipment should be designed to operate continuously for several hours a day in the warmest months.

EXAMPLE

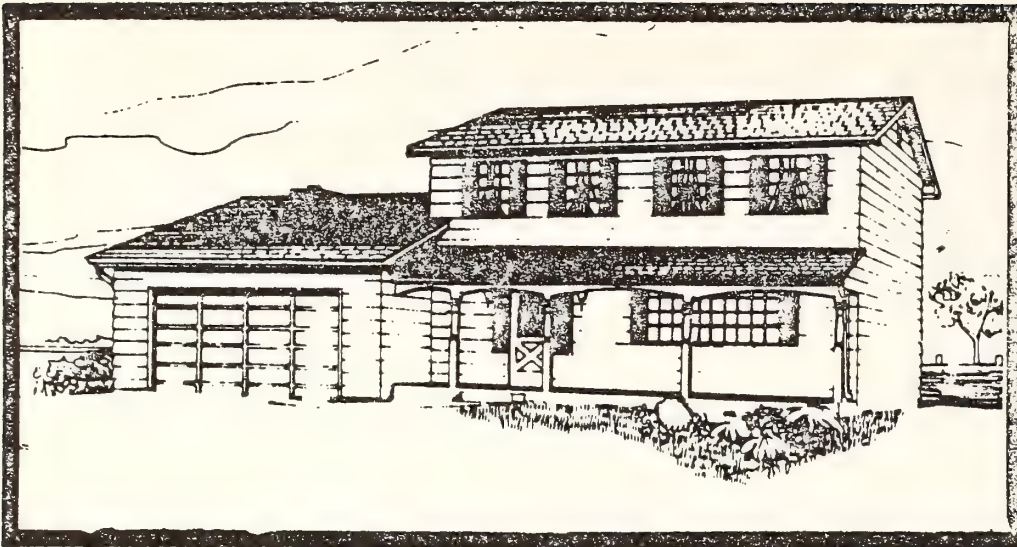
The cooling load for the house of Figure 5-1 is calculated as shown in the Cooling Load Worksheet, pages 5-27 and 5-28. The outdoor design temperature is 89°F (not listed in Table A5-10 for Fort Collins). The indoor design temperature is 75°F. The U factors for walls, ceiling and door are the same as for winter conditions. Refinements in U factors were not made in these computations although the R factors in air films in Tables A5-1 and A5-2 would result in slightly different R factors.

The overhangs over the south-facing windows effectively reduce the heat transfer rates to values equivalent to the north-facing windows, and there are no east- and west-facing windows. No credit is taken for shades or drapes over the windows.

The temperature in the garage was assumed to be the mean between indoor and outdoor design temperatures, and the design equivalent temperature differences (DETD) given in Table A5-7 were interpolated for the design outdoor temperature of 89°F.

The total cooling load for the building is calculated to be 18,621 Btuh. This low cooling load is a result of low design outdoor temperature in Fort Collins, 89°F, and a building which is insulated properly with shading over windows. The values used apply for average summer conditions, so on days when temperatures reach 95°F, greater cooling

capacity is required. If not provided, interior temperature will rise above the desired setting. But if the air-conditioner is sized for the outdoor design temperature of 89°F, it will operate continuously in hot weather, and temperature excursions inside the building should not be large.



TOTAL FLOOR AREA
 2078 ft^2 - 2 flo
 plus 1182 ft^2 - Base
 3260 ft^2

Colonial two-story with all the necessary size and luxury for a large or growing family • Four Bedrooms and Two Baths on the Second Floor • Large Entry With Open Stairway • Spacious Living Room • Formal Dining Room • U-Shape Kitchen With Eating Space • Family Room With Fireplace Located Next To Kitchen • Full Unfinished Basement • Two Car Garage • Paneling

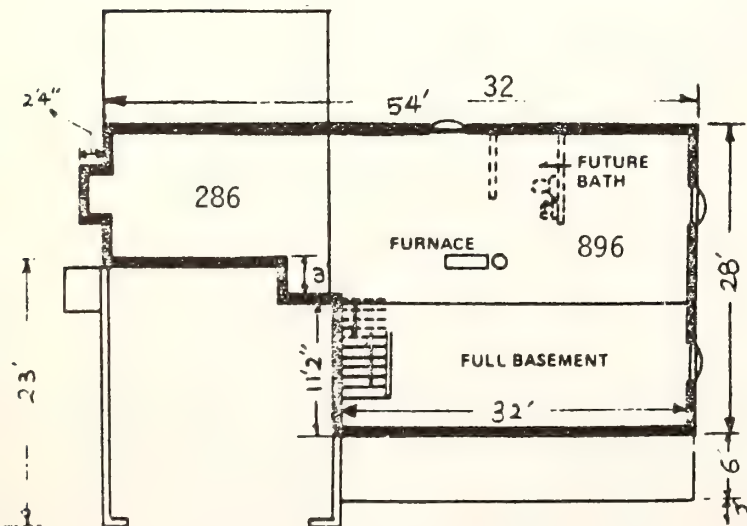
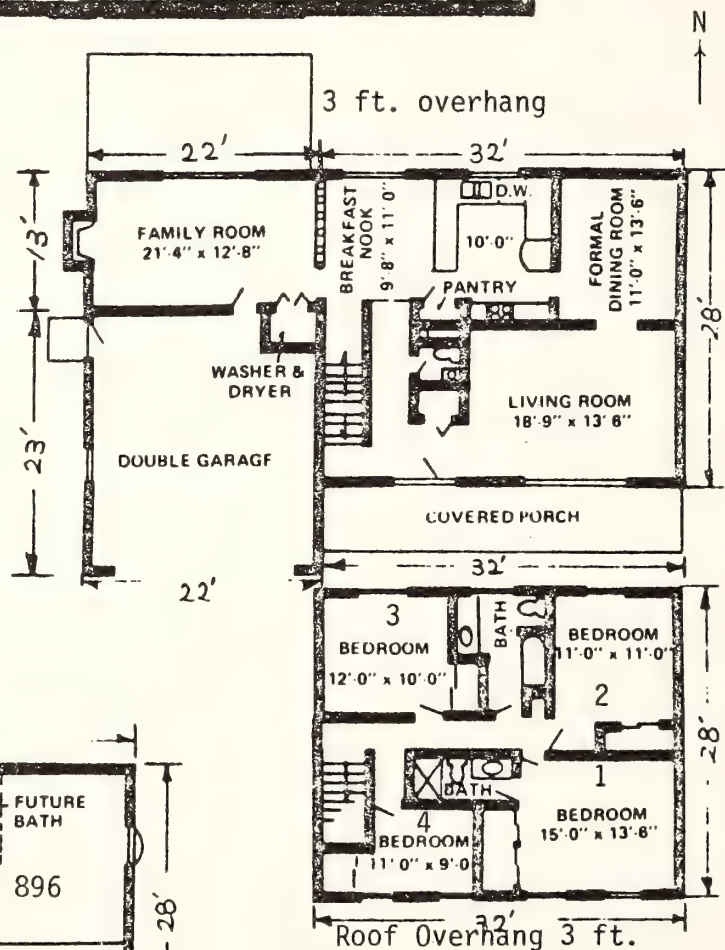


Figure 5-1. Example Residential Building

PLAN 809

Table 5-1

FHA Form 2005
VA Form 26-1857
Rev. 2/74

U. S. DEPARTMENT OF HOUSING AND URBAN DEVELOPMENT
FEDERAL HOUSING ADMINISTRATION

For accurate register of carbon copies, form
may be separated along above fold. Sample
completed sheets together in original order.

Form Approved
OMB No. 68-00033

☐ Proposed Construction

DESCRIPTION OF MATERIALS No. _____

(To be inserted by FHA or VA)

☐ Under Construction

Property address _____ City _____ State _____

Mortgagor or Sponsor _____ (Name) _____ (Address)

Contractor or Builder Barran Homes, Inc. _____ (Name) _____ (Address)

INSTRUCTIONS

1. For additional information on how this form is to be submitted, number of copies, etc., see the instructions applicable to the FHA Application for Mortgage Insurance or VA Request for Determination of Reasonable Value, as the case may be.

2. Describe all materials and equipment to be used, whether or not shown on the drawings, by marking an X in each appropriate check-box and entering the information called for in each space. If space is inadequate, enter "See spec." and describe under item 27 or on an attached sheet. THE USE OF PAINT CONTAINING MORE THAN FIVE-TENTHS OF ONE PERCENT LEAD BY WEIGHT IS PROHIBITED.

3. Work not specifically described or shown will not be considered

unless required, then the minimum acceptable will be assumed. Work exceeding minimum requirements cannot be considered unless specifically described.

4. Include no alternates, "or equal" phrases, or contradictory items. (Consideration of a request for acceptance of substitute materials or equipment is not thereby precluded.)

5. Include signatures required at the end of this form.

6. The construction shall be completed in compliance with the related drawings and specifications, as amended during processing. The specifications include this Description of Materials and applicable Minimum Property Standards.

1. EXCAVATION:

Bearing soil, type Clay

2. FOUNDATIONS:

Footings: concrete mix 5 sack; strength psi 2500#; Reinforcing none

Foundation wall: material concrete; Reinforcing see plans

Interior foundation wall: material concrete; Party foundation wall see plans

Columns: material and sizes see plans; Piers: material and reinforcing see plans

Olders: material and sizes see plans; Sills: material see plans

Basement entrance airway see plans; Window airways see plans

Waterproofing hot spray asphalt; Footing drains none

Termite protection _____

Basementless space: ground cover 55# felt; insulation _____; foundation vents bird screen

Special foundations _____

Additional information: _____

3. CHIMNEYS:

Material metal; Prefabricated (make and size) _____

Flue lining: material metal; Heater flue size 8"; Fireplace flue size 9" I.D.

Vents (material and size): gas or oil heater _____; water heater _____

Additional information: _____

4. FIREPLACES:

Type: ☒ solid fuel; ☐ gas-burning; ☐ circulator (make and size) _____; Ash dump and clean-out _____

Fireplace: facing Brick Veneer; lining metal; hearth brick; mantel wood

Additional information: Heatilator Mark 123 Model 3036

5. EXTERIOR WALLS:

Wood frame: wood grade, and species W.C. const. fir; ☐ Corner bracing. Building paper or felt _____

Sheathing insul board; thickness 1/2"; width _____; ☒ solid; ☐ spaced _____" o. c.; ☐ diagonal;

Siding WOOD; grade _____; type _____; size _____; exposure _____; fastening galv. nails _____

Shingles _____; grade _____; type _____; size _____; exposure _____; fastening galv. nails _____

Bricks _____; thickness _____; Lath _____; weight _____ lb.

Masonry veneer _____; Sills _____; Lintels none; Base flashing metal

Masonry: ☐ solid ☐ faced ☐ stuccoed; total wall thickness _____; facing thickness _____; facing material _____

Backup material _____; thickness _____; bonding _____

Door sills _____; Window sills _____; Lintels _____; Base flashing _____

Interior surfaces: dampproofing, _____ coats of _____; furring _____

Additional information: _____

Exterior painting: material Jones Blair exterior paint; number of coats 2

Cable wall construction: ☐ same as main walls, ☐ other construction _____

6. FLOOR FRAMING:

Joists: wood, grade, and species W.C. const. fir; other _____; bridging 1x3; anchors 1/2x10

Concrete slab: ☒ basement floor, ☐ first floor, ☐ ground supported; ☐ self-supporting; mix 5 sack; thickness 8"

reinforcing 6/6-10/10 WCF; insulation _____; membrane polyurethane

Fill under slab: material gravel; thickness 4"; Additional information: _____

7. SUBFLOORING: (Describe underflooring for special floors under item 21.)

Material: grade and species 3/4" tung and groove; size 4x8; type CG

Laid: ☐ first floor, ☐ second floor, ☐ attic _____; sq. ft. ☐ diagonal; ☒ right angles. Additional information: _____

8. FINISH FLOORING: (Wood only. Describe other finish flooring under item 21.)

Location _____ Rooms _____ Color _____ Species _____ Thickness _____ Width _____ Base _____ Finish _____

First floor _____

Second floor _____

Attic floor _____ sq. ft. _____

Additional information: _____

Table 5-1 (continued)

DESCRIPTION OF MATERIALS

9. PARTITION FRAMING:

Bruds: wood, grade, and species W.C. cove fir size and spacing 2x4 @ 24" o.c. Other: _____
 Additional information: Bearing Walls: 2x4 @ 16" o.c.

10. CEILING FRAMING:

Joists: wood, grade, and species _____ Other _____ Bridging _____
 Additional information: truss (see attached detail)

11. ROOF FRAMING:

Rafters: wood, grade, and species _____ Roof trusses (see detail): grade and species 45° pitch
 Additional information: truss (see attached detail)

12. ROOFING:

Sheathing: wood, grade, and species 1/2" C.D. plywood ; ☒ solid; ☐ spaced _____" o.c.
 Roofing Asphalt ; grade 235d ; size _____ ; type _____
 Underlay Felt ; weight or thickness 15 ; size _____ ; fastening galv. nails
 Built-up roofing _____ ; number of plies _____ ; surfacing material _____
 Flashing: material galv. metal ; gage or weight 30 ; ☐ gravel stops; ☐ snow guards
 Additional information: _____

13. GUTTERS AND DOWNSPOUTS:

Gutters: material galv. ; gage or weight 26 ; size 12 ; shape rounded
 Downspouts: material galv. ; gage or weight 26 ; size 3 ; shape square ; number 1
 Downspouts connected to: ☐ Storm sewer; ☐ sanitary sewer; ☐ dry-well; ☒ Splash blocks: material and size concrete
 Additional information: _____

14. LATH AND PLASTER

Lath ☐ walls; ☐ ceilings: material _____ ; weight or thickness _____ Plaster: coats _____ ; finish _____
 Dry-wall ☐ walls; ☒ ceilings: material gyp. bd. ; thickness 1/2 ; finish texture
 Joint treatment tape

15. DECORATING: (Paint, wallpaper, etc.)

Rooms	Wall Finish Material and Application	Ceiling Finish Material and Application
Kitchen	(1) prime & (2) enamel coats	same
Bath	"	"
Other	(1) coat rubber base	"

Additional information: applies only to finished areas (see plans)

16. INTERIOR DOORS AND TRIM:

Doors: type wood flush ; material mahogany ; thickness 1 3/8"
 Door trim: type S. line ; material white pine Base: type S. line ; material white pine ; size 5 1/4"
 Finish: doors Fill (2) coats stain ; trim (1) prime, (2) enamel coats
 Other trim (item, type and location) window sills: formica
 Additional information: closet doors: metal bi-fold/louvered

17. WINDOWS:

General Aluminum Corp. TARTAN TEL. 321-4316
 Windows: type sliding ; make Series 1800 ; material aluminum ; sash thickness _____
 Glass: grade insulated ; ☐ sash weights; ☐ balances, type _____ ; head flashing _____
 Trim: type _____ ; material _____ Paint _____ ; number coats _____
 Weatherstripping: type _____ ; material _____ Storm sash, number _____
 Screens: ☐ full; ☒ half; type _____ ; number all ; screen cloth material galv.
 Basement windows: type sliding ; material aluminum ; screens, number all ; Storm sash, number _____
 Special windows _____
 Additional information: _____

18. ENTRANCES AND EXTERIOR DETAIL:

Main entrance door: material Mahogany ; width 36" ; thickness 1 3/8" Frame: material fir ; thickness 1 3/8"
 Other entrance doors: material Fir ; width 32" ; thickness 1 3/8" Frame: material " ; thickness "
 Head flashing galv. metal Weatherstripping: type _____ ; saddles _____
 Screen doors: thickness 1" ; number 1 ; screen cloth material galv. Storm doors: thickness _____ ; number _____
 Combination storm and screen doors: thickness 1" ; number 1 ; screen cloth material galv.
 Shutters: ☐ hinged; ☐ fixed. Railings _____ , Arlic louvers _____
 Exterior millwork: grade and species _____ Paint _____ ; number coats _____
 Additional information: _____

19. CABINETS AND INTERIOR DETAIL:

Manufactured by Alpine cabinet Co. see
 Kitchen cabinets, wall units: material Timnath, Colo. ; linear feet of shelves plans ; shelf width 12"
 Base units: material _____ ; counter top formica ; edging same
 Back and end splash formica Finish of cabinets wood grain vinyl ; number coats _____
 Medicine cabinets: make _____ ; model _____
 Other cabinets and built-in furniture bath vanities per plans
 Additional information: _____

20. STAIRS:

STAIR	TREADS		RISERS		STRINGERS		HANDRAIL		BALUSTERS	
	Material	Thickness	Material	Thickness	Material	Size	Material	Size	Material	Size
Basement	fir	5/4	pine	3/4	W.C. fir	2x12	wood	2"	W.I.	3/4"
Main										
Attic										

Disappearing: make and model number _____

Additional information: _____

Table 5-1 (continued)

21. SPECIAL FLOORS AND WAINSCOT:

	Location	Material, Color, Border, Size, Gage, Etc.	Threshold Matings	Wall Base Material	Underfloor Matings
Floors	Kitchen	Armstrong or equal		rubber	plywood
	Bath	"		"	"
	entry	"		"	"
	other	Carpet (finished areas only) (see attached)		pine	"
Wainscot	Location	Material, Color, Border, Cap, Size, Gage, Etc.	Height	Height Over Tls	Height in Showers (From Floor)
	Bath	Ceramic tile	72"	63"	72"

Bathroom accessories: ☒ Recessed; material chrome; number bath; ☐ Attached; material chrome; number bath
 Additional information: _____

22. PLUMBING:

Fixture	Number	Location	Make	Unit's Fixture Identification No.	Size	Color
Sink	1	kitchen	Briggs	3401	21x32	white
Lavatory	1	bath	Amur Standard	S1007-056	19" Dia.	"
Water closet	1	"	Kohler	R3512-PB		"
Bathrub	1	"	Briggs	3000	30x60	"
Shower over tub Δ	1	"				
Stall shower Δ						
Laundry trays						

Δ ☒ Curtain rod ☐ Door ☐ Shower pan: material _____
 Water supply: ☒ public; ☐ community system; ☐ individual (private) system. \star
 Sewage disposal: ☒ public; ☐ community system; ☐ individual (private) system. \star
 \star State and describe individual system in complete detail in separate drawings and specifications according to requirements.
 House drain (inside): ☐ cast iron; ☐ tile; ☒ other ABS House sewer (outside): ☐ cast iron; ☒ tile; ☐ other _____
 Water piping: ☐ galvanized steel; ☒ copper tubing; ☐ other _____ Sill cocks, number 2
 Domestic water heater: type Gas; make and model A. O. Smith; heating capacity 42,000 BTU
35-3 gph. 100° rise. Storage tank: material glass lined; capacity 40 gallons.
 Gas service: ☒ utility company; ☐ liq. pet. gas; ☐ other _____ Gas piping: ☐ cooking; ☐ house heating.
 Footing drains connected to: ☐ storm sewer; ☐ sanitary sewer; ☐ dry well. Sump pump; make and model _____; capacity _____; discharges 1/2 in.

23. HEATING:

☒ Hot water ☐ Steam ☐ Vapor ☐ One-pipe system ☒ Two-pipe system.
☐ Radiators ☐ Convectors ☐ Baseboard radiation. Make and model _____
 Radiant panel: ☐ floor; ☐ wall; ☐ ceiling. Panel coil: material _____
☐ Circulator ☐ Return pump. Make and model _____; capacity _____ gpm.
 Boiler: make and model _____ Output _____ Btu/h; net rating _____ Btu/h.
 Additional information: _____
 Warm air: ☐ Gravity ☒ Forced. Type of system perimeter 120-150 BTU.
 Duct material: supply galv.; return galv. Insulation _____, thickness _____ ☐ Outside air intake.
 Furnace: make and model Lennox Input see plans Btu/h; output see plans Btu/h.
 Additional information: _____
☐ Space heater; ☐ floor furnace; ☐ wall heater. Input _____ Btu/h; output _____ Btu/h; number units _____
 Make, model _____ Additional information: _____
 Controls: make and type Lennox
 Additional information: _____
 Fuel: ☐ Coal; ☐ oil; ☐ gas; ☐ liq. pet. gas; ☐ electric; ☐ other _____; storage capacity _____
 Additional information: _____
 Firing equipment furnished separately: ☐ Gas burner, conversion type. ☐ Stoker: hopper feed ☐ bin feed ☐
 Oil burner: ☐ pressure atomizing; ☐ vaporizing _____
 Make and model _____ Control _____
 Additional information: _____
 Electric heating system: type _____ Input _____ watts; @ _____ volts; output _____ Btu/h.
 Additional information: _____
 Ventilating equipment: attic fan, make and model _____; capacity _____ cfm.
 kitchen exhaust fan, make and model see item # 26
 Other heating, ventilating, or cooling equipment: _____

24. ELECTRIC WIRING:

Service: ☐ overhead; ☒ underground. Panel: ☐ fuse box; ☒ circuit-breaker; make Klinco AMP's 100 No. circuits 15
 Wiring: ☐ conduit; ☐ armored cable; ☒ nonmetallic cable; ☐ knob and tube; ☐ other _____
 Special outlets: ☒ range; ☐ water heater; ☒ other dryer
☒ Doorbell ☒ Chimes. Push-button locations front door Additional information: _____

25. LIGHTING FIXTURES:

Total number of fixtures see plans Total allowance for fixtures, typical installation, $\$$ 150.00
 Nontypical installation _____
 Additional information: _____

DESCRIPTION OF MATERIALS

Table 5-1 (continued)

26. INSULATION:

DESCRIPTION OF MATERIALS

LOCATION	THICKNESS	MATERIAL, TYPE, AND METHOD OF INSTALLATION	VAPOR BARRIER
Roof			
Ceiling	6"	blown rock wool	R-10
Wall	3 1/2"	batt	R-11
Floor			
			482-9059
			482-3181

HARDWARE: (make, material, and finish.) Stanley, brass, smooth

Privacy lock at master bedroom and bathrooms; keyed locks at all entrance doors including garage doors; all other doors passage knobs.

SPECIAL EQUIPMENT: (State material or make, model and quantity. Include only equipment and appliances which are acceptable by local law, custom and applicable FHA standards. Do not include items which, by established custom, are supplied by occupant and removed when he vacates premises or chattles prohibited by law from becoming realty.)

Garbage Disposal - Insinkerator Badger

Dishwasher - Frigidaire DW 3CDU1

Range - " RBE 353

Hood - Nautilus

Optional fireplace - Heatilator Mark 123 Model 3036

Optional Medicine Cabinet - Recessed Kent Model WAL 1420

27. MISCELLANEOUS: (Describe any main dwelling materials, equipment, or construction items not shown elsewhere; or use to provide additional information where the space provided was inadequate. Always reference by item number to correspond to numbering used on this form.)

Provide hot & cold water for washer

" 110 outlet for washer

" 220 outlet for dryer

PORCHES:

see plans

TERRACES:

see plans

GARAGES:

attached see plans

WALKS AND DRIVEWAYS:

Driveway: width 17'; base material gravel; thickness 4"; surfacing material concrete; thickness 4"

Front walk: width 3'; material concrete; thickness 4". Service walk: width _____; material _____; thickness _____

Steps: material concrete; treads 12 1/2"; risers 6 1/2". Check walls _____

OTHER ONSITE IMPROVEMENTS:

(Specify all exterior onsite improvements not described elsewhere, including items such as unusual grading, drainage structures, retaining walls, fence, railings, and accessory structures.)

LANDSCAPING, PLANTING, AND FINISH GRADING:

Topsoil: 4" thick: ☐ front yard; ☐ side yards; ☐ rear yard to 0-11 feet behind main building.

Lawns (seeded, sodded, or sprigged): ☐ front yard sodded; ☐ side yards _____; ☐ rear yard _____

Planting: ☐ as specified and shown on drawings; ☐ as follows:

_____ Shade trees, deciduous, _____" caliper.

_____ Low flowering trees, deciduous, _____' to _____'

_____ High-growing shrubs, deciduous, _____' to _____'

_____ Medium-growing shrubs, deciduous, _____' to _____'

_____ Low-growing shrubs, deciduous, _____' to _____'

_____ Evergreen trees, _____' to _____', B & B

_____ Evergreen shrubs, _____' to _____', B & B

_____ Vines, 2-year _____

Identification.—This exhibit shall be identified by the signature of the builder, or sponsor, and/or the proposed mortgagor if the latter is known at the time of application.

Date 21 January 1975

Signature _____

WORKSHEET FOR HEAT LOAD CALCULATIONS
(for Example Building)

BUILDING SECTION	SIZE OR VOLUME	NET AREA OR VOLUME	U COEFF.	TEMP. DIFF. [68-(-9)]	HEAT LOSS *	TOTALS *
BEDROOM 1						
South wall	(15+3)x8	120	.07	77	647	
East wall	13.5x8	108	.07	77	282	
Windows (2)	3x4	24	.50	77	924	
Infiltration	2/3x15x13.6x8	1088	.018	77	1508	3361
BEDROOM 2						
East wall	14x8	112	.07	77	604	
North wall	11x8	76	.07	77	410	
Window	3x4	12	.50	77	462	
Infiltration	2/3x11x11x8	645	.018	77	894	2370
BATHROOM						
North wall	8x8	60	.07	77	323	
Window	2x2	4	.50	77	154	
Infiltration	3/4x7.5x11x8	495	.018	77	686	1163
BEDROOM 3						
North wall	12x8	84	.07	77	453	
West wall	10x8	80	.07	77	431	
Window	3x4	12	.50	77	462	
Infiltration	2/3x10x12x8	640	.018	77	887	2233
BEDROOM 4 & HALLWAY						
West wall	16x8	128	.07	77	690	
South wall	14x8	88	.07	77	474	
Windows (2)	3x4	24	.50	77	924	
Infiltration	2/3x14x16x8	1195	.018	77	1656	3744
LIVING ROOM						
South wall	32x8	203	.07	77	1094	
Door	3x7	21	.26	77	420	
Window	4x8	32	.62	77	1528	
East wall	13.5x8	108	.07	77	582	
Infiltration	2/3x19x13.5x8	1368	.018	77	1896	5520
DINING ROOM						
East wall	13.5x8	108	.07	77	582	
North wall	11x8	88	.07	77	474	
Infiltration	1/3x11x13.5x8	396	.018	77	549	1650

WORKSHEET FOR HEAT LOAD CALCULATIONS
(for Example Building)

BUILDING SECTION	SIZE OR VOLUME	NET AREA OR VOLUME	U COEFF.	TEMP. DIFF. [68-(-9)]	HEAT LOSS *	TOTALS *
KITCHEN, BREAKFAST						
North wall	18x8	122	.07	77	657	
Window	2.5x4	10	.50	77	385	
Window	3.4	12	.50	77	462	
Infiltration	1x18x11x8	1584	.018	77	2195	3699
FAMILY ROOM						
North wall	21.5x8	136	.07	77	733	
Patio Door	6x6	36	.58	77	1608	
West wall	13x8	104	.20	77	1602	
South wall	22x8	176	.52	38	3478	
Infiltration	2x13x22x8	4576	.018	77	6342	13763
HALL						
West wall	17x8	136	.52	38	2687	
Infiltration	1x8x8x17	1088	.018	77	1508	4195
BASEMENT						
North wall	54x8	432	.10	23	994	
West wall	28x8	224	.10	23	515	
South wall	54x8	432	.10	23	994	
East wall	28x8	224	.10	23	515	
Floor	32x28	896	.10	23	2061	
Floor	13x22	286	.10	23	658	
Infiltration	1/6x54x13x8+ 1/6x15x32x8	1576	.018	77	2184	7921
CEILING						
Second floor	32x28	896	.04	77	2760	
Family room	13x22	286	.04	77	681	3641

*Btuh

TOTAL HEAT LOSS RATE (Btuh) 53215

$$\text{Unit Heating Load} = \frac{53215}{68-(-9)} \times 24 = 16590 \text{ Btu/DD}$$

$$\text{Annual Heating Load} = 16590 \times 6283 = 104.2 \text{ MMBtu}$$

COOLING LOAD WORKSHEET
(for Example Building)

BUILDING SECTION	SIZE OR VOLUME	NET AREA OR VOLUME	U or UNIT HEAT GAIN	DETD	HEAT GAIN	TOTALS
BEDROOM 1						
South wall	18x8	120	.07	19	160	
East wall	13.5x8	108	.07	19	144	
Windows (2)	3x4	24	.27		648	
Infiltration	1632	816	.018	14	205	1157
BEDROOM 2						
East wall	14x8	112	.07	19	149	
North wall	11x8	76	.07	19	101	
Window	3x4	12	.27		324	
Infiltration	968	484	.018	14	122	696
BATHROOM						
North wall	8x8	60	.07	19	80	
Window	2x2	4	.27		108	
Infiltration	660	660	.018	14	166	354
BEDROOM 3						
North wall	12x8	84	.07	19	112	
West wall	10x8	80	.07	19	106	
Window	3x4	12	.27		324	
Infiltration	960	480	.018	14	121	663
BEDROOM 4 AND HALLWAY						
West wall	16x8	128	.07	19	170	
South wall	14x8	88	.07	19	117	
Windows (2)	3x4	24	.27		648	
Infiltration	1792	896	.018	14	226	1161
LIVING ROOM						
South wall	32x8	203	.07	19	270	
Door	3x7	21	.47	19	188	
Window	4x8	32	.21		672	
East Wall	13.5x8	108	.15	11	178	
Infiltration	2052	1026	.018	14	258	1566
DINING ROOM						
East wall	13.5x8	108	.07	19	144	
North wall	11x8	88	.07	19	117	
Infiltration	1188	198	.018	14	50	311

COOLING LOAD WORKSHEET
(for Example Building)

BUILDING SECTION	SIZE OR VOLUME	NET AREA OR VOLUME	U or UNIT HEAT GAIN	DETD	HEAT GAIN	TOTALS
KITCHEN, BREAKFAST						
North wall	1848	122	.07	19	162	
Windows (2)		22	.27		594	
Infiltration	1584	1584	.918	14	399	1155
FAMILY ROOM						
North wall	21.5x8	136	.07	19	180	
West wall	13x8	104	.20	19	395	
South wall	22x8	176	.52	7	640	
Patio door	6x6	36	.21		756	
Infiltration	2288	2288	.018	14	577	2548
HALL						
West wall	17x8	136	.52	7	495	
Infiltration	1088	1088	.018	14	274	769
CEILING						
Second floor	32x28	896	.04	39	1398	
Family room	13x22	286	.04	39	446	1844

TOTAL

12224

No load is calculated for
basement. No credit for
cool basement taken.

4 occupants x 225	900
Kitchen appliances	1200
Total Sensible Heat Gain	<u>14324</u>
Latent Heat Gain (30%x14324)	4297
Latent + Sensible Heat Gain	<u>18621</u>
Cooling Load, Btuh	18621

APPENDIX

Table A5-1

Surface Conductances and Resistances for Air Films

ITEMS	WINTER		SUMMER	
	f	R	f	R
INTERIOR SURFACES				
Ceiling	1.63	0.61	1.08	0.92*
Sloped ceiling 45°	1.60	0.62	1.32	0.76*
Walls and windows	1.46	0.68	1.46	0.68
Floor	1.08	0.92	1.08	0.92
EXTERIOR SURFACES				
Roofs, walls and windows	6.00	0.17+	4.00	0.25 [†]

* Heat flow direction reversed from winter conditions

+ 15 mph wind

[†] 7.5 mph wind

Table A5-2

Resistance Values for Air Spaces

ITEM	Air Space	WINTER		SUMMER	
		3/4"	4"	3/4"	4"
Flat roof		1.02	1.12	0.87	0.94
Wall		1.28	1.16	1.01	1.01

Table A5-4

U Factors for Windows and Patio Doors Btu/(hr·ft²·°F)

DESCRIPTION	WINTER
SINGLE GLASS	
Metal sash	1.13
Wood sash, 80% glass	0.02
DOUBLE GLASS:	
1/4" Air Space	
Metal sash	0.65
Wood sash, 80% glass	0.62
Wood sash, 60% glass	0.55
1/2" Air Space	
Metal sash	0.70
Wood sash, 80% glass	0.49
TRIPLE GLASS	
1/4" Air Space	
Metal sash	0.56
Wood sash, 80% glass	0.45
STORM WINDOWS	
1" to 4" Air Space	
Wood	0.50
Metal	0.56
SLIDING PATIO DOORS	
Single Glass	
Wood frame	1.07
Metal frame	1.13
Double Glass, 1/2" Air Space	
Wood frame	0.58
Metal frame	0.64

Table A5-5

U Factors for Solid Doors Btu/(hr·ft²·°F)

THICKNESS (IN)	WINTER			SUMMER WITHOUT STORM DOOR
	WITHOUT STORM DOOR	WITH STORM DOOR, 50% GLASS		
		WOOD	METAL	
1	0.64	0.30	0.39	0.61
1 ¼	0.55	0.28	0.34	0.53
1 ½	0.49	0.27	0.33	0.47
2	0.43	0.24	0.29	0.42

Table A5-6

Air Changes for Average Residential Conditions

KIND OF ROOM	AIR CHANGE PER HOUR	
	WINTER	SUMMER
Rooms with no windows or exterior doors	1/3	1/6
Rooms with windows or exterior doors on one side	2/3	1/2
Rooms with windows or exterior doors on two sides	1	2/3
Rooms with windows or exterior doors on three sides	1 1/3	1
Entrance halls and air locks	1 1/2	1

Table A5-7

Design Equivalent Temperature Differences (°F)

DESIGN OUTDOOR TEMPERATURE	85	95		105
TEMPERATURE RANGE DURING DAY	15-25	15-25	>25	>25
WALLS AND DOORS				
Wood frame and doors	14	24	19	29
Masonry	6	16	11	21
CEILINGS AND ROOF				
Under vented attic, dark roof	34	44	39	49
Built-up roof (no ceiling), light roof	26	36	31	41
FLOORS				
Over unconditioned rooms and open crawl space	5	15	10	20
Over basement, enclosed crawl space	0	0	0	0

Table A5-8
Design Heat Gains Through Windows Btu/(hr·ft²)

OUTDOOR DESIGN TEMPERATURE	SINGLE PANE			DOUBLE PANE		
	85	95	105	85	95	105
NO AWNINGS OR INSIDE SHADING						
North	23	31	38	19	24	28
Northeast; Northwest	56	64	71	46	51	55
East and West	81	89	96	68	73	77
Southeast; Southwest	70	78	85	59	64	68
South	40	48	55	33	38	42
WITH DRAPERIES OR VENETIAN BLINDS						
North	15	23	30	12	17	21
Northeast; Northwest	32	40	47	27	32	36
East and West	48	56	63	42	47	51
Southeast; Southwest	40	48	55	35	40	44
South	23	31	38	20	25	29
ROLLER SHADES, HALF DOWN						
North	18	26	33	15	20	24
Northeast, Northwest	40	48	55	38	43	47
East and West	61	69	76	54	59	63
Southeast; Southwest	52	60	67	46	51	55
South	29	37	44	26	32	36
AWNINGS						
North	20	28	35	13	18	22
Northeast; Northwest	21	29	36	14	19	23
East and West	22	30	37	14	19	23
Southeast; Southwest	21	29	36	14	19	23
South	21	28	35	13	18	22

Table A5-9

Shade Line Factors* (5 hour average, 1 August)

WINDOW ORIENTATION	LATITUDE					
	25	30	35	40	45	50
East and West	0.8	0.8	0.8	0.8	0.8	0.8
Southeast; Southwest	1.9	1.6	1.4	1.3	1.1	1.0
South	10.1	5.4	3.6	2.6	2.0	1.7

*Multiply shade line factors by width of overhang to determine shadow line below overhang.

Table A5-10

Heating Degree Days* and Design Outdoor Temperature

NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)															Design T _o °F	
STATE AND STATION	JULY	AUG.	SEPT	OCT.	NOV.	DEC.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	ANNUAL	WIN [†]	SUM ^{II}	
ALABAMA																
Birmingham	0	0	6	93	363	555	592	462	363	108	9	0	2551	19	97	
Huntsville	0	0	12	127	426	663	694	557	434	138	19	0	3070	13	97	
Mobile	0	0	0	22	213	357	415	300	211	42	0	0	1560	26	95	
Montgomery	0	0	0	68	330	527	543	417	316	90	0	0	2291	22	98	
ALASKA																
Anchorage	245	291	516	930	1284	1572	1631	1316	1293	879	592	315	10864	-25	73	
Annette	242	208	327	567	738	899	949	837	843	648	490	321	7069			
Barrow	803	840	1035	1500	1971	2362	2517	2332	2468	1944	1445	957	20174	-45	58	
Barter Island	735	775	987	1482	1944	2337	2536	2369	2477	1923	1373	924	19862			
Bethel	319	394	612	1042	1434	1866	1903	1590	1655	1173	806	402	13196			
Cold Bay	474	425	525	772	918	1122	1153	1036	1122	951	792	591	9880			
Cordova	366	391	522	781	1017	1221	1299	1086	1113	864	660	444	9764			
Fairbanks	171	332	642	1203	1833	2254	2359	1901	1739	1068	555	222	14279	-53	82	
Juneau	301	338	483	725	921	1135	1237	1070	1073	810	601	381	9705	-7	75	
King Salmon	313	322	513	908	1290	1606	1600	1333	1411	966	673	408	11343			
Kotzebue	381	446	723	1249	1728	2127	2192	1932	2080	1554	1057	636	16105			
McGrath	208	338	633	1184	1791	2232	2294	1817	1758	1122	648	258	14283			
Nome	481	496	693	1904	1455	1820	1879	1666	1770	1314	930	573	14171	-32	66	
Saint Paul	605	539	612	862	963	1197	1228	1168	1265	1098	936	726	11199			
Shemya	577	475	501	784	876	1042	1045	958	1011	885	837	696	9687			
Yakutat	338	347	474	716	936	1144	1169	1019	1042	840	632	435	9092			

* From Climatic Atlas of the United States, U. S. Department of Commerce, Env. Sci. Serv. Adm. June, 1968

+ From Table 1, Chapter 33, ASHRAE Handbook of Fundamentals 1972 (99% of time warmer than this temperature)

II From Table 1, Chapter 33, ASHRAE Handbook of Fundamentals 1972 (1% of time dry bulb temperature is greater)

Table A5-10 (continued)

NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)															Design T _o °F	
STATE AND STATION	JULY	AUG.	SEPT	OCT.	NOV.	DEC.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	ANNUAL	WIN [†]	SUM ^{II}	
ARIZONA																
Flagstaff	46	68	201	558	867	1073	1169	991	911	651	437	180	7152	0	84	
Phoenix	0	0	0	22	234	415	474	328	217	75	0	0	1765	31	108	
Prescott	0	0	27	245	579	797	865	711	605	360	158	15	4362	15	96	
Tucson	0	0	0	25	231	406	471	344	242	75	6	0	1800	29	105	
Winslow	0	0	6	245	711	1008	1054	770	601	291	96	0	4782	9	97	
Yuma	0	0	0	0	148	319	363	228	130	29	0	0	1217	37	111	
ARKANSAS																
Fort Smith	0	0	12	127	450	704	781	591	456	144	22	0	3292	15	101	
Little Rock	0	0	9	127	465	716	756	577	434	126	9	0	3219	19	99	
Texarkana	0	0	0	78	345	561	626	468	350	105	0	0	2533	22	99	
CALIFORNIA																
Bakersfield	0	0	0	37	282	502	546	364	267	105	19	0	2122	31	103	
Bishop	0	0	42	248	576	797	874	666	539	306	143	36	4227			
Blue Canyon	34	50	120	347	579	766	865	781	791	582	397	195	5507			
Burbank	0	0	6	43	177	301	366	277	239	138	81	18	1646	36	97	
Eureka	270	257	258	329	414	499	546	470	505	438	372	285	4643	32	67	
Fresno	0	0	0	78	339	558	586	406	319	150	56	0	2492	28	101	
Long Beach	0	0	12	40	156	288	375	297	267	168	90	18	1711	36	87	
Los Angeles	28	22	42	78	180	291	372	302	288	219	158	81	2061	42	94	
Mt. Shasta	25	34	123	406	696	902	983	784	738	525	347	159	5722			
Oakland	53	50	45	127	309	481	527	400	353	255	180	90	2870	35	85	
Point Arguello	202	186	162	205	291	400	474	392	403	339	298	243	3595			
Red Bluff	0	0	0	53	318	555	605	428	341	168	47	0	2515			
Sacramento	0	0	12	81	363	577	614	442	360	216	102	6	2773	30	100	
Sandberg	0	0	30	202	480	691	778	661	620	426	264	57	4209			
San Diego	6	0	15	37	123	251	313	249	202	123	84	36	1439	42	86	
San Francisco	81	78	60	143	306	462	508	395	363	279	214	126	3015	42	80	
Santa Catalina	16	0	9	50	165	279	353	308	326	249	192	105	2052			
Santa Maria	99	93	96	146	270	391	459	370	363	282	233	165	2967	32	85	

Table A5-10 (continued)

NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)															Design T _o °F	
STATE AND STATION	JULY	AUG.	SEPT	OCT.	NOV.	DEC.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	ANNUAL	WIN [†]	SUM ^{II}	
COLORADO																
Alamosa	65	99	279	639	1065	1420	1476	1162	1020	696	440	168	8529	-17	84	
Colorado Springs	9	25	132	456	825	1032	1128	938	893	582	319	84	6423	-1	90	
Denver	6	9	117	428	819	1035	1132	938	887	558	288	66	6283	-2	92	
Grand Junction	0	0	30	313	786	1113	1209	907	729	387	146	21	5641	8	96	
Pueblo	0	0	54	326	750	986	1085	871	772	429	174	15	5462	-5	96	
CONNECTICUT																
Bridgeport	0	0	66	307	615	986	1079	966	853	510	208	27	5617	4	90	
Hartford	0	6	99	372	711	1119	1209	1061	899	495	177	24	6172	1	90	
New Haven	0	12	87	347	648	1011	1097	991	871	543	245	45	5897	5	88	
DELAWARE																
Wilmington	0	0	51	270	588	927	980	874	735	387	112	6	4930	12	93	
FLORIDA																
Apalachicola	0	0	0	16	153	319	347	260	180	33	0	0	1308	32	94	
Daytona Beach	0	0	0	0	75	211	248	190	140	15	0	0	879	38	94	
Fort Myers	0	0	0	0	24	109	146	101	62	0	0	0	442	29	96	
Jacksonville	0	0	0	12	144	310	332	246	174	21	0	0	1239	55	90	
Key West	0	0	0	0	0	28	40	31	9	0	0	0	108	35	95	
Lakeland	0	0	0	0	57	164	195	146	99	0	0	0	661	45	91	
Miami Beach	0	0	0	0	0	40	56	36	9	0	0	0	141	33	96	
Orlando	0	0	0	0	72	198	220	165	105	6	0	0	766	29	92	
Pensacola	0	0	0	19	195	353	400	277	183	36	0	0	1463	25	96	
Tallahassee	0	0	0	28	198	360	375	286	202	36	0	0	1485	36	92	
Tampa	0	0	0	0	60	171	202	148	102	0	0	0	683	40	92	
West Palm Beach	0	0	0	0	6	65	87	64	31	0	0	0	253	40	92	

Table A5-10 (continued)

NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)															Design T _o °F	
STATE AND STATION	JULY	AUG.	SEPT	OCT.	NOV.	DEC.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	ANNUAL	WIN [†]	SUM ^{II}	
<u>GEORGIA</u>																
Athens	0	0	12	115	405	632	642	529	431	141	22	0	2929	17	96	
Atlanta	0	0	18	127	414	626	639	529	437	168	25	0	2983	18	95	
Augusta	0	0	0	78	333	552	549	445	350	90	0	0	2397	20	98	
Columbus	0	0	0	87	333	543	552	434	338	96	0	0	2383	23	98	
Macon	0	0	0	71	297	502	505	403	295	63	0	0	2136	23	98	
Rome	0	0	24	161	474	701	710	577	468	177	34	0	3326	16	97	
Savannah	0	0	0	47	24	437	437	353	254	45	0	0	1819	24	96	
Thomasville	0	0	0	25	198	366	394	305	208	33	0	0	1529			
<u>IDAHO</u>																
Boise	0	0	132	415	792	1017	1113	854	722	438	245	81	5809	4	96	
Idaho Falls 46W	16	34	270	623	1056	1370	1538	1249	1085	651	391	192	8475			
Idaho Falls 42NW	16	40	282	648	1107	1432	1600	1291	1107	657	388	192	8760			
Lewiston	0	0	123	403	756	933	1063	815	694	426	239	90	5542	6	98	
Pocatello	0	0	172	493	900	1166	1324	1058	905	555	319	141	7033	-8	94	
<u>ILLINOIS</u>																
Cairo	0	0	36	164	513	791	856	680	539	195	47	0	3821	-3	94	
Chicago	0	0	81	326	753	1113	1209	1044	890	480	211	48	6155	-7	94	
Moline	0	9	99	335	774	1181	1314	1100	918	450	189	39	6408	-2	94	
Peoria	0	6	87	326	759	1113	1218	1025	849	426	183	33	6025	-7	92	
Rockford	6	9	114	400	837	1221	1333	1137	961	516	236	60	6830	-1	95	
Springfield	0	0	72	291	696	1023	1135	935	769	354	136	18	5429			
<u>INDIANA</u>																
Evansville	0	0	66	220	606	896	955	767	620	237	68	0	4435	6	96	
Fort Wayne	0	9	105	378	783	1135	1178	1028	890	471	189	39	6205	0	93	
Indianapolis	0	0	90	316	723	1051	1113	949	809	432	177	39	5699	0	93	
South Bend	0	6	111	372	777	1125	1221	1070	933	525	239	60	6439	-2	92	

Table A5-10 (continued)

NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)															Design T _o °F	
STATE AND STATION	JULY	AUG.	SEPT	OCT.	NOV.	DEC.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	ANNUAL	WIN [†]	SUM ^{II}	
<u>IOWA</u>																
Burlington	0	0	93	322	768	1135	1259	1042	859	426	177	33	6144	-4	95	
Des Moines	0	9	99	363	837	1231	1398	1163	967	489	211	39	6808	-7	95	
Dubuque	12	31	156	450	906	1287	1420	1204	1026	546	260	78	7376	-11	92	
Sioux City	0	9	108	369	867	1240	1435	1198	989	483	214	39	6951	-10	96	
Waterloo	12	19	138	428	909	1296	1460	1221	1023	531	229	54	7320	-12	91	
<u>KANSAS</u>																
Concordia	0	0	57	276	705	1023	1163	935	781	372	149	18	5479			
Dodge City	0	0	33	251	666	939	1051	840	719	354	124	9	4968	3	99	
Goodland	0	6	81	381	810	1073	1166	955	884	507	236	42	6141	-2	99	
Topeka	0	0	57	270	672	980	1122	893	722	330	124	12	5182	3	99	
Wichita	0	0	33	229	618	905	1023	804	645	270	87	6	4620	5	102	
<u>KENTUCKY</u>																
Covington	0	0	75	291	669	983	1035	893	756	390	149	24	5264	3	93	
Lexington	0	0	54	239	609	902	946	818	685	325	105	0	4683	6	94	
Louisville	0	0	54	248	609	890	930	818	682	315	105	9	4660	8	96	
<u>LOUISIANA</u>																
Alexandria	0	0	0	56	273	431	471	361	260	69	0	0	1921	25	97	
Baton Rouge	0	0	0	31	216	369	409	294	208	33	0	0	1560	25	96	
Burrwood	0	0	0	0	96	214	298	218	171	27	0	0	1024			
Lake Charles	0	0	0	19	210	341	381	274	195	39	0	0	1459	29	95	
New Orleans	0	0	0	19	192	322	363	258	192	39	0	0	1385	32	93	
Shreveport	0	0	0	47	297	477	552	426	304	81	0	0	2184	22	99	
<u>MAINE</u>																
Caribou	78	115	336	682	1044	1535	1690	1470	1308	858	468	183	9767	-18	85	
Portland	12	53	195	508	807	1215	1339	1182	1042	675	372	111	7511	-5	88	

Table A5-10 (continued)

NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)															
STATE AND STATION	JULY	AUG.	SEPT	OCT.	NOV.	DEC.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	ANNUAL	WIN [†]	SUM [‡]
<u>MARYLAND</u>															
Baltimore	0	0	48	264	585	905	936	820	679	327	90	0	4654	16	94
Frederick	0	0	66	307	624	955	995	876	741	384	127	12	5087	7	94
<u>MASSACHUSETTS</u>															
Blue Hill Obsy	0	22	108	381	690	1085	1178	1053	936	579	267	69	6368		
Boston	0	9	60	316	603	983	1088	972	846	513	208	36	5634	6	91
Nantucket	12	22	93	332	573	896	992	941	896	621	384	129	5891		
Pittsfield	25	59	219	524	831	1231	1339	1196	1063	660	326	105	7578	-1	86
Worcester	6	34	147	450	774	1172	1271	1123	998	612	304	78	6969	1	89
<u>MICHIGAN</u>															
Alpena	68	105	273	580	912	1268	1404	1299	1218	777	446	156	8506	-5	87
Detroit (City)	0	0	87	360	738	1088	1181	1058	936	522	220	42	6232	4	92
Escanaba	59	87	243	539	924	1293	1445	1296	1203	777	456	159	8481	-7	82
Flint	16	40	159	465	843	1212	1330	1198	1066	639	319	90	7377	-1	89
Grand Rapids	9	28	135	434	804	1147	1259	1134	1011	549	279	75	6894	2	91
Lansing	6	22	138	431	813	1163	1262	1142	1011	579	273	69	6909	2	89
Marquette	59	81	240	527	936	1268	1411	1268	1187	771	468	177	8393	-8	88
Muskegon	12	28	120	400	762	1088	1209	1100	995	594	319	78	6696	4	87
Sault Ste. Marie	96	105	279	580	951	1367	1525	1380	1277	810	477	201	9048	-12	83
<u>MINNESOTA</u>															
Duluth	71	109	330	632	1131	1581	1745	1518	1355	840	490	198	10000	-19	85
International Falls	71	112	363	701	1236	1724	1919	1621	1414	828	443	174	10606	-29	86
Minneapolis	22	31	189	505	1014	1454	1631	1380	1166	621	288	81	8382	-14	92
Rochester	25	34	186	474	1005	1438	1593	1366	1150	630	301	93	8295	-17	90
Saint Cloud	28	47	225	549	1065	1500	1702	1445	1221	666	326	105	8879	-20	90
<u>MISSISSIPPI</u>															
Jackson	0	0	0	65	315	502	546	414	310	87	0	0	2239	21	98
Meridian	0	0	0	81	339	518	543	417	310	81	0	0	2289	20	97
Vicksburg	0	0	0	53	279	462	512	384	282	69	0	0	2041	23	97

Table A5-10 (continued)

NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)															Design T _o °F
STATE AND STATION	JULY	AUG.	SEPT	OCT.	NOV.	DEC.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	ANNUAL	WIN [†]	SUM ^{II}
<u>MISSOURI</u>															
Columbia	0	0	54	251	651	967	1076	874	716	324	121	12	5046	2	97
Kansas	0	0	39	220	612	905	1032	818	682	294	109	0	4711	4	100
St. Joseph	0	6	60	285	708	1039	1172	949	769	348	133	15	5484	-1	97
St. Louis	0	0	60	251	627	936	1026	848	704	312	121	15	4900	7	96
Springfield	0	0	45	223	600	877	973	781	660	291	105	6	4561	5	97
<u>MONTANA</u>															
Billings	6	15	186	487	897	1135	1296	1100	970	570	285	102	7049	-10	94
Glasgow	31	47	270	608	1104	1466	1711	1439	1187	648	335	150	8996	-25	96
Great Falls	28	53	258	543	921	1169	1349	1154	1063	642	384	186	7750	-20	91
Havre	28	53	306	595	1065	1367	1584	1364	1181	657	338	162	8700	-22	91
Helena	31	59	294	601	1002	1265	1438	1170	1042	651	381	195	8129	-17	90
Kalispell	50	99	321	654	1020	1240	1401	1134	1029	639	397	207	8191	-7	88
Miles City	6	6	174	502	972	1296	1504	1252	1057	579	276	99	7723	-19	97
Missoula	34	74	303	651	1035	1287	1420	1120	970	621	391	219	8125	-7	92
<u>NEBRASKA</u>															
Grand Island	0	6	108	381	834	1172	1314	1089	980	462	211	45	6530	-6	98
Lincoln	0	6	75	301	726	1066	1237	1016	834	402	171	30	5864	-4	100
Norfolk	9	0	111	397	873	1234	1414	1179	983	498	233	48	6979	-11	97
North Platte	0	6	123	440	895	1166	1271	1039	930	519	248	57	6684	-6	97
Omaha	0	12	105	357	828	1175	1355	1126	939	465	208	42	6612	-5	97
Scottsbluff	0	0	138	459	876	1128	1231	1008	921	552	285	75	6673	-8	96
Valentine	9	12	165	493	942	1237	1395	1176	1045	579	288	84	7425		
<u>NEVADA</u>															
Elko	9	34	225	561	924	1197	1314	1036	911	621	409	192	7433	-13	94
Ely	28	43	234	592	939	1184	1308	1075	977	672	456	225	7733	-6	90
Las Vegas	0	0	0	78	387	617	688	487	335	111	6	0	2709	23	108
Reno	43	87	204	490	801	1026	1073	823	729	510	357	189	6332	12	94
Winnemucca	0	34	210	536	876	1091	1172	916	837	573	363	153	6761	1	97

Table A5-10 (continued)

NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)															Design T _o °F	
STATE AND STATION	JULY	AUG.	SEPT	OCT.	NOV.	DEC.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	ANNUAL	WIN [†]	SUM [‡]	
<u>NEW HAMPSHIRE</u>																
Concord	6	50	177	505	822	1240	1358	1184	1032	636	298	75	7383	-11	91	
Mt. Wash. Obsy.	493	536	720	1057	1341	1742	1820	1663	1652	1260	930	603	13817			
<u>NEW JERSEY</u>																
Atlantic City	0	0	39	251	549	880	936	848	741	420	133	15	4812	14	91	
Newark	0	0	30	248	573	921	983	876	729	381	118	0	4859	11	94	
Trenton	0	0	57	264	576	924	989	885	753	399	121	12	4980	12	92	
<u>NEW MEXICO</u>																
Albuquerque	0	0	12	229	642	868	930	703	595	288	81	0	4348	14	96	
Clayton	0	6	66	310	699	899	986	812	747	429	183	21	5158			
Raton	9	28	126	431	825	1048	1116	904	834	543	301	63	6228	-2	92	
Roswell	0	0	18	202	573	806	840	641	481	201	31	0	3793	16	101	
Silver City	0	0	6	183	525	729	791	605	581	261	87	0	3705	14	95	
<u>NEW YORK</u>																
Albany	0	19	138	440	777	1194	1311	1156	992	564	239	45	6875	1	91	
Binghamton (AP)	22	65	201	471	810	1184	1277	1154	1045	645	313	99	7286	-2	91	
Binghamton (PO)	0	28	141	406	732	1107	1190	1081	949	543	229	45	6451			
Buffalo	19	37	141	440	777	1156	1256	1145	1039	645	329	78	7062	-5	90	
Central Park	0	0	30	233	540	902	986	885	760	408	118	9	4871	11	94	
JF Kennedy Intl.	0	0	36	248	564	933	1029	935	815	480	167	12	5219	17	91	
La Guardia	0	0	27	223	528	887	973	879	750	414	124	6	4811	12	93	
Rochester	9	31	126	415	747	1125	1234	1123	1014	597	279	48	6748	2	91	
Schenectady	0	22	123	422	756	1159	1283	1131	970	543	211	30	6650	-5	90	
Syracuse	6	28	132	415	744	1153	1271	1140	1004	570	248	45	6756	-2	90	

Table A5-10 (continued)

NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)															Design T _o °F	
STATE AND STATION	JULY	AUG.	SEPT	OCT.	NOV.	DEC.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	ANNUAL	WIN [†]	SUM [‡]	
<u>NORTH CAROLINA</u>																
Asheville	0	0	48	245	555	775	784	683	592	273	87	0	4042	13	91	
Cape Hatteras	0	0	0	78	273	521	580	518	440	177	25	0	2612		96	
Charlotte	0	0	6	124	438	691	691	582	481	156	22	0	3191	18		
Greensboro	0	0	33	192	513	778	784	672	552	234	47	0	3805	14	94	
Raleigh	0	0	21	164	450	716	725	616	487	180	34	0	3393	16	95	
Wilmington	0	0	0	74	291	521	546	462	357	96	0	0	2347	23	94	
Winston Salem	0	0	21	171	483	747	753	652	524	207	37	0	3595	14	94	
<u>NORTH DAKOTA</u>																
Bismarck	34	28	222	577	1083	1463	1708	1442	1203	645	329	117	8851	-24	95	
Devils Lake	40	53	273	642	1191	1634	1872	1579	1345	753	381	138	9901	-23	93	
Fargo	28	37	219	574	1107	1569	1789	1520	1262	690	332	99	9226	-22	92	
Williston	31	43	261	601	1122	1513	1758	1473	1263	681	357	141	9243	-21	94	
<u>OHIO</u>																
Akron	0	9	96	381	726	1070	1138	1016	871	489	202	39	6037	1	89	
Cincinnati	0	0	54	248	612	921	970	837	701	336	118	9	4806	8	94	
Cleveland	9	25	105	384	738	1088	1159	1047	918	552	260	66	6351	2	91	
Columbus	0	6	84	347	714	1039	1088	949	809	426	171	27	5660	2	92	
Dayton	0	6	78	310	696	1045	1097	955	809	429	167	30	5622	0	92	
Mansfield	9	22	114	397	768	1110	1169	1042	924	543	245	60	6403	1	91	
Sandusky	0	6	66	313	684	1032	1107	991	868	495	198	36	5796	4	92	
Toledo	0	16	117	406	792	1138	1200	1056	924	543	242	60	6494	1	92	
Youngstown	6	19	120	412	771	1104	1169	1047	921	540	248	60	6417	1	89	
<u>OKLAHOMA</u>																
Oklahoma City	0	0	15	164	498	766	868	664	527	189	34	0	3725	11	100	
Tulsa	0	0	18	158	522	787	893	683	539	213	47	0	3860	12	102	

Table A5-10 (continued)

NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)															Design T _o °F	
STATE AND STATION	JULY	AUG.	SEPT	OCT.	NOV.	DEC.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	ANNUAL	WIN [†]	SUM [‡]	
OREGON																
Astoria	146	130	210	375	561	679	753	622	636	480	363	231	5186	27	79	
Burns	12	37	210	515	867	1113	1246	988	856	570	366	177	6957			
Eugene	34	34	129	366	585	719	803	627	589	426	279	135	4726	22	91	
Meacham	84	124	288	580	918	1091	1209	1005	983	726	527	339	7874			
Medford	0	0	78	372	678	871	918	697	642	432	242	78	5008	21	98	
Pendleton	0	0	111	350	711	884	1017	773	617	396	205	63	5127	3	97	
Portland	25	28	114	335	597	735	825	644	586	396	245	105	3635	26	91	
Roseburg	22	16	105	329	567	713	766	608	570	405	267	123	4491	25	93	
Salem	37	31	111	338	594	729	822	647	611	417	273	144	4754	21	92	
Sexton Summit	81	81	171	443	666	874	958	809	818	609	465	279	6524			
PENNSYLVANIA																
Allentown	0	0	90	353	693	1045	1116	1002	849	471	167	24	5810	3	92	
Erie	0	25	102	391	714	1063	1169	1081	973	585	288	60	6451	7	88	
Harrisburg	0	0	63	298	648	992	1045	907	766	396	124	12	5251	9	92	
Philadelphia	0	0	60	291	621	964	1014	890	744	390	115	12	5101	11	93	
Pittsburgh	0	9	105	375	726	1063	1119	1002	874	480	195	39	5987	7	90	
Reading	0	0	54	257	597	939	1001	885	735	372	105	0	4945	6	92	
Scranton	0	19	132	434	762	1104	1156	1028	893	498	195	33	6254	2	89	
Willimasport	0	9	111	375	717	1073	1122	1002	856	468	177	24	5934	1	91	
RHODE ISLAND																
Block Island	0	16	78	307	594	902	1020	955	877	612	344	99	5804			
Providence	0	16	96	372	660	1023	1110	988	868	534	236	51	5954	6	89	
SOUTH CAROLINA																
Charleston	0	0	0	59	282	471	487	389	291	54	0	0	2033	26	95	
Columbia	0	0	0	84	345	577	570	470	357	81	0	0	2484	20	98	
Florence	0	0	0	78	315	552	552	459	347	84	0	0	2387	21	96	
Greenville	0	0	0	112	387	636	648	535	434	120	12	0	2884	19	95	
Spartanburg	0	0	15	130	417	667	663	560	453	144	25	0	3074	18	95	

Table A5-10 (continued)

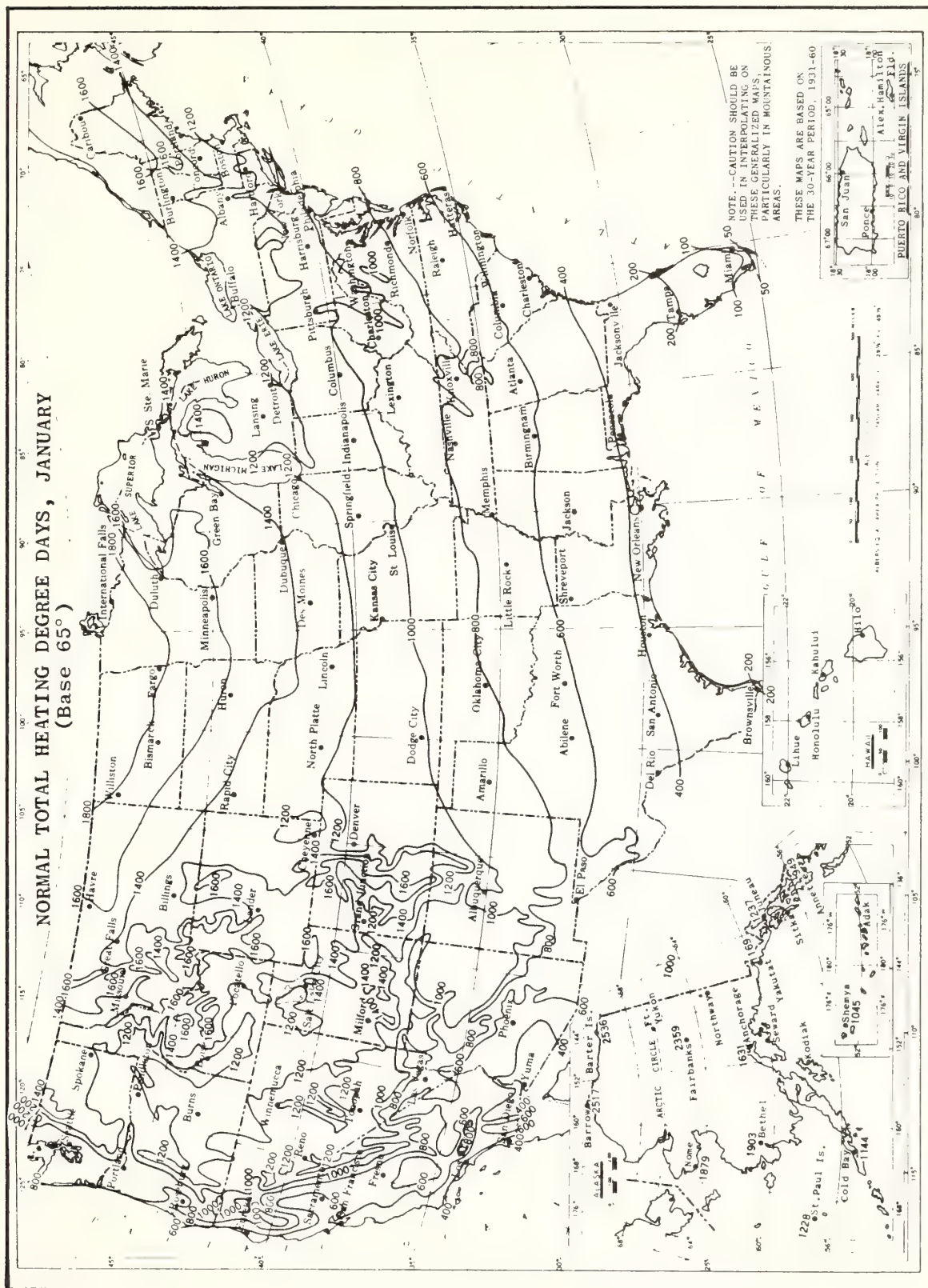
NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)															Design T _o °F
STATE AND STATION	JULY	AUG.	SEPT	OCT.	NOV.	DEC.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	ANNUAL	WIN [†]	SUM [‡]
<u>SOUTH DAKOTA</u>															
Huron	9	12	165	508	1014	1432	1628	1355	1125	600	288	87	8223	-16	97
Rapid City	22	12	165	481	897	1172	1333	1145	1051	615	326	126	7345	-9	96
Sioux Falls	19	25	168	462	972	1361	1544	1285	1082	573	270	78	7839	-14	95
<u>TENNESSEE</u>															
Bristol	0	0	51	236	573	828	828	700	598	261	68	0	4143	11	92
Chattanooga	0	0	18	143	468	698	722	577	453	150	25	0	3254	15	97
Knoxville	0	0	30	171	489	725	732	613	493	198	43	0	3494	13	95
Memphis	0	0	18	130	447	698	729	585	456	147	22	0	3232	17	98
Nashville	0	0	30	158	495	732	778	644	512	189	40	0	3578	12	97
Oak Ridge (CO)	0	0	39	192	531	772	778	669	552	228	56	0	3817		
<u>TEXAS</u>															
Abilene	0	0	0	99	366	586	642	470	347	114	0	0	2624	17	101
Amarillo	0	0	18	205	570	797	877	664	546	252	56	0	3985	8	98
Austin	0	0	0	31	225	388	468	325	223	51	0	0	1711	25	101
Brownsville	0	0	0	0	66	149	205	106	74	0	0	0	600	36	94
Corpus Christi	0	0	0	0	120	220	291	174	109	0	0	0	914	32	95
Dallas	0	0	0	62	321	524	601	440	319	90	6	0	2363	19	101
El Paso	0	0	0	84	414	648	685	445	319	105	0	0	2700	21	100
Fort Worth	0	0	0	65	324	536	614	448	319	99	0	0	2405	20	102
Galveston	0	0	0	0	138	270	350	258	189	30	0	0	1235	32	91
Houston	0	0	0	6	183	307	384	288	192	36	0	0	1396	29	96
Laredo	0	0	0	0	105	217	267	134	74	0	0	0	797	32	103
Lubbock	0	0	18	174	513	744	800	613	484	201	31	0	3578	11	99
Midland	0	0	0	87	381	592	651	468	322	90	0	0	2591	19	100
Port Arthur	0	0	0	22	207	329	384	274	192	39	0	0	1447	29	94
San Angelo	0	0	0	68	318	536	567	412	288	66	0	0	2255	20	101
San Antonio	0	0	0	31	207	363	428	286	195	39	0	0	1549	25	99
Victoria	0	0	0	6	150	270	344	230	152	21	0	0	1173	28	98
Waco	0	0	0	43	270	456	536	389	270	66	0	0	2030	21	101
Wichita Falls	0	0	0	99	381	632	698	518	378	120	6	0	2832	15	103

Table A5-10 (continued)

NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)															Design T _O °F	
STATE AND STATION	JULY	AUG.	SEPT	OCT.	NOV.	DEC.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	ANNUAL	WIN [†]	SUM ^{II}	
<u>UTAH</u> Milford Salt Lake City Wendover	0	0	99	443	867	1141	1252	988	822	519	279	87	6497			
	0	0	81	419	849	1082	1172	910	763	459	233	84	6052	5	97	
	0	0	48	372	822	1091	1178	902	729	408	177	51	4778			
<u>VERMONT</u> Burlington	28	65	207	539	891	1349	1513	1333	1187	714	353	90	8269	-12	88	
<u>VIRGINIA</u> Cape Henry Lynchburg Norfolk Richmond Roanoke Wash. Nat'l. AP	0	0	0	112	360	645	694	633	536	246	53	0	3279			
	0	0	51	223	540	822	849	731	605	267	78	0	4166	15	94	
	0	0	0	136	408	698	738	655	533	216	37	0	3421	20	94	
	0	0	36	214	495	784	815	703	546	219	53	0	3865	14	96	
	0	0	51	229	549	825	834	722	614	261	65	0	4150	15	94	
	0	0	33	217	519	834	871	762	626	288	74	0	4224			
<u>WASHINGTON</u> Olympia Seattle Seattle Boeing Seattle Tacoma Spokane Stampede Pass Tatoosh Island Walla Walla Yakima	68	71	198	422	636	753	834	675	645	450	307	177	5326	21	85	
	50	47	129	329	543	657	738	599	577	396	242	177	4424	23	82	
	34	40	147	384	624	763	831	655	608	411	242	99	4838			
	56	62	162	391	633	750	828	678	657	474	295	159	5145	20	85	
	9	25	168	493	879	1082	1231	980	834	531	288	135	6655	-2	93	
	273	291	393	701	1008	1178	1287	1075	1085	855	654	483	9283			
	295	279	306	406	534	639	713	613	645	525	431	333	5719			
	0	0	87	310	681	843	986	745	589	342	177	45	4805	12	98	
	0	12	144	450	828	1039	1163	868	713	435	220	69	5941	6	94	
<u>WEST VIRGINIA</u> Charleston Elkins Huntington Parkersburg	0	0	63	254	591	865	880	770	648	330	96	9	4476	9	92	
	9	25	135	400	729	992	1008	896	791	444	198	48	5675	1	87	
	0	0	63	257	585	856	880	764	636	294	99	12	4446	10	95	
	0	0	60	264	606	905	942	826	691	339	115	6	4754	8	93	

Table A5-10 (continued)

NORMAL TOTAL HEATING DEGREE DAYS (Base 65°)														Design T _o °F	
STATE AND STATION	JULY	AUG.	SEPT.	OCT.	NOV.	DEC.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	ANNUAL	WIN [†]	SUM [‡]
<u>WISCONSIN</u>															
Green Bay	28	50	174	484	924	1333	1494	1313	1141	654	335	99	8029	-12	88
La Crosse	12	19	153	437	924	1139	1504	1277	1070	540	245	69	7589	-12	90
Madison	25	40	174	474	930	1330	1473	1274	1113	618	310	102	7863	-9	92
Milwaukee	43	47	174	471	876	1252	1376	1193	1054	642	372	135	7635	-6	90
<u>WYOMING</u>															
Casper	6	16	192	524	942	1169	1290	1084	1020	657	381	129	7410	-11	92
Cheyenne	19	31	210	543	924	1101	1228	1056	1011	672	381	102	7278	-6	89
Lander	6	19	204	555	1020	1299	1417	1145	1017	654	381	153	7870	-16	92
Sheridan	25	31	219	538	948	1200	1355	1154	1054	642	366	150	7683	-12	95



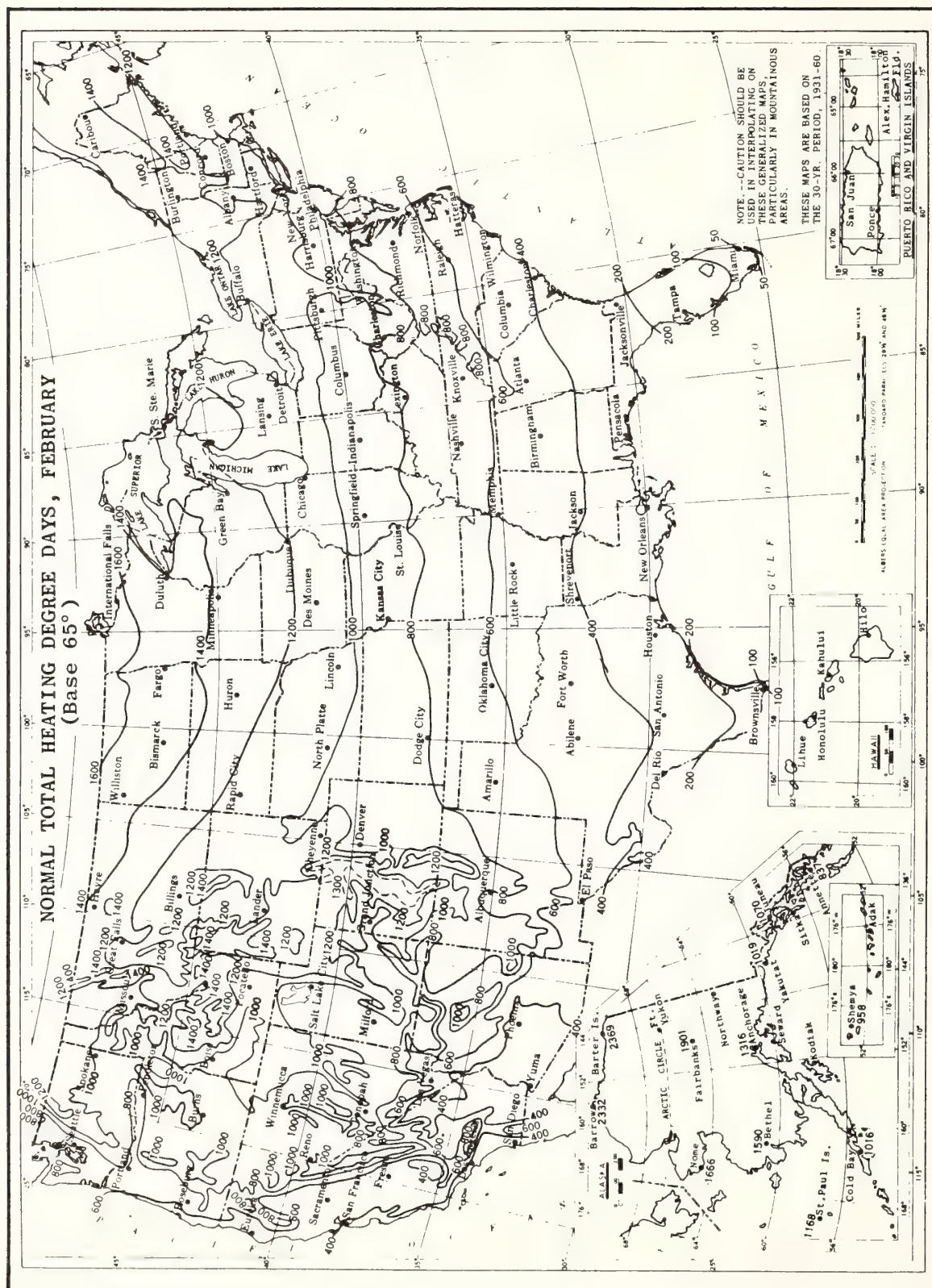
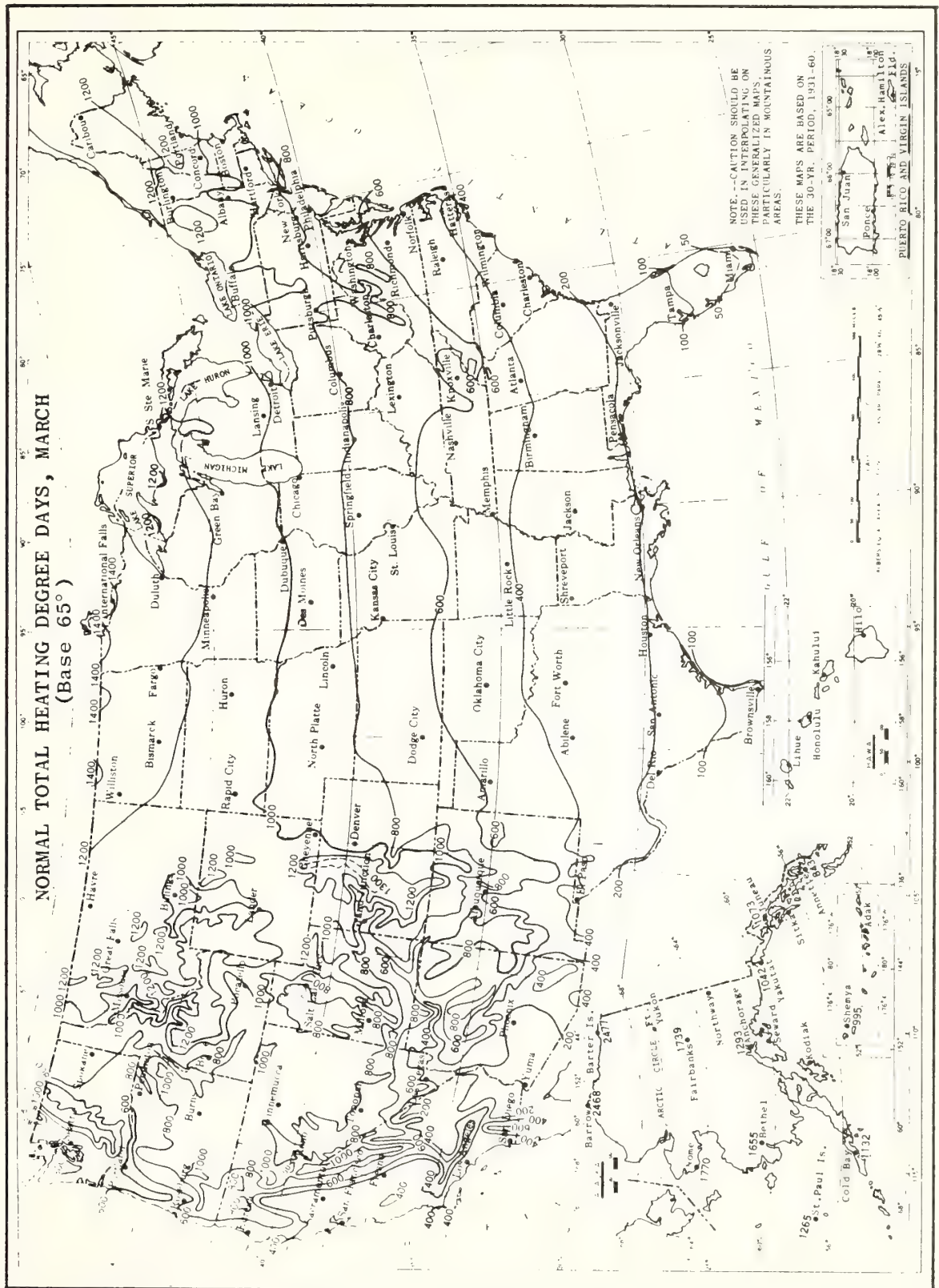
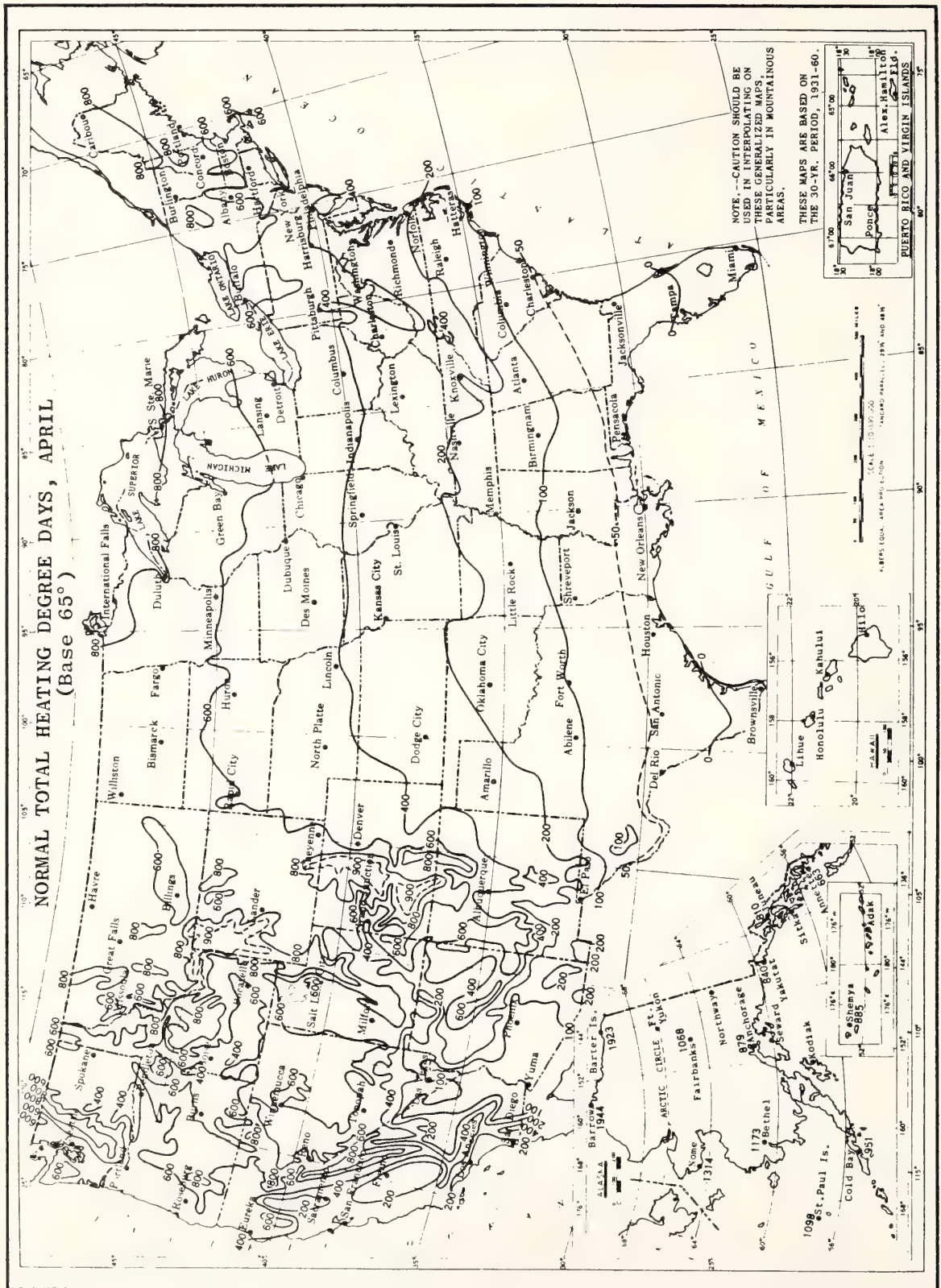


Figure A5-3





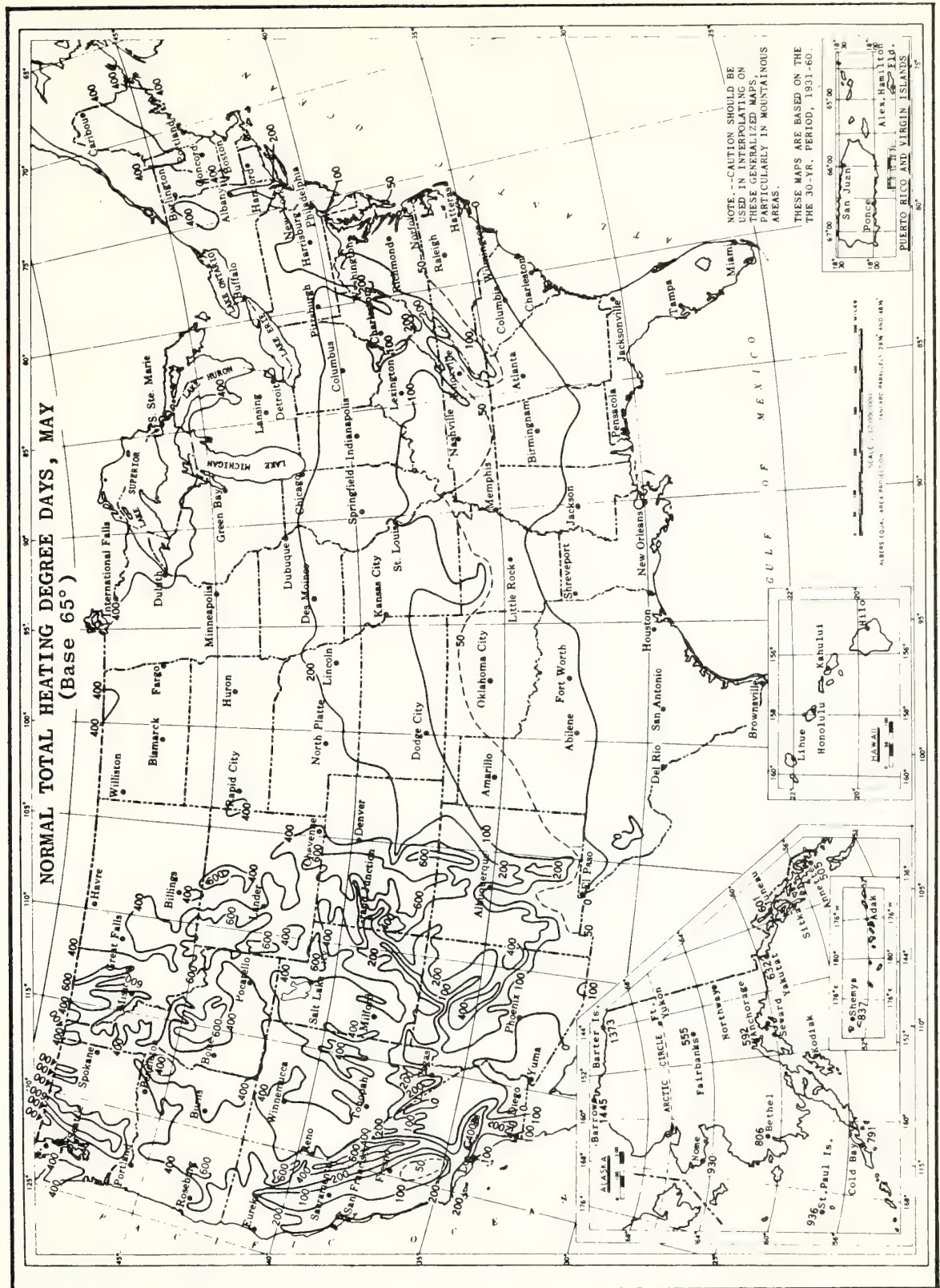


Figure A5-6

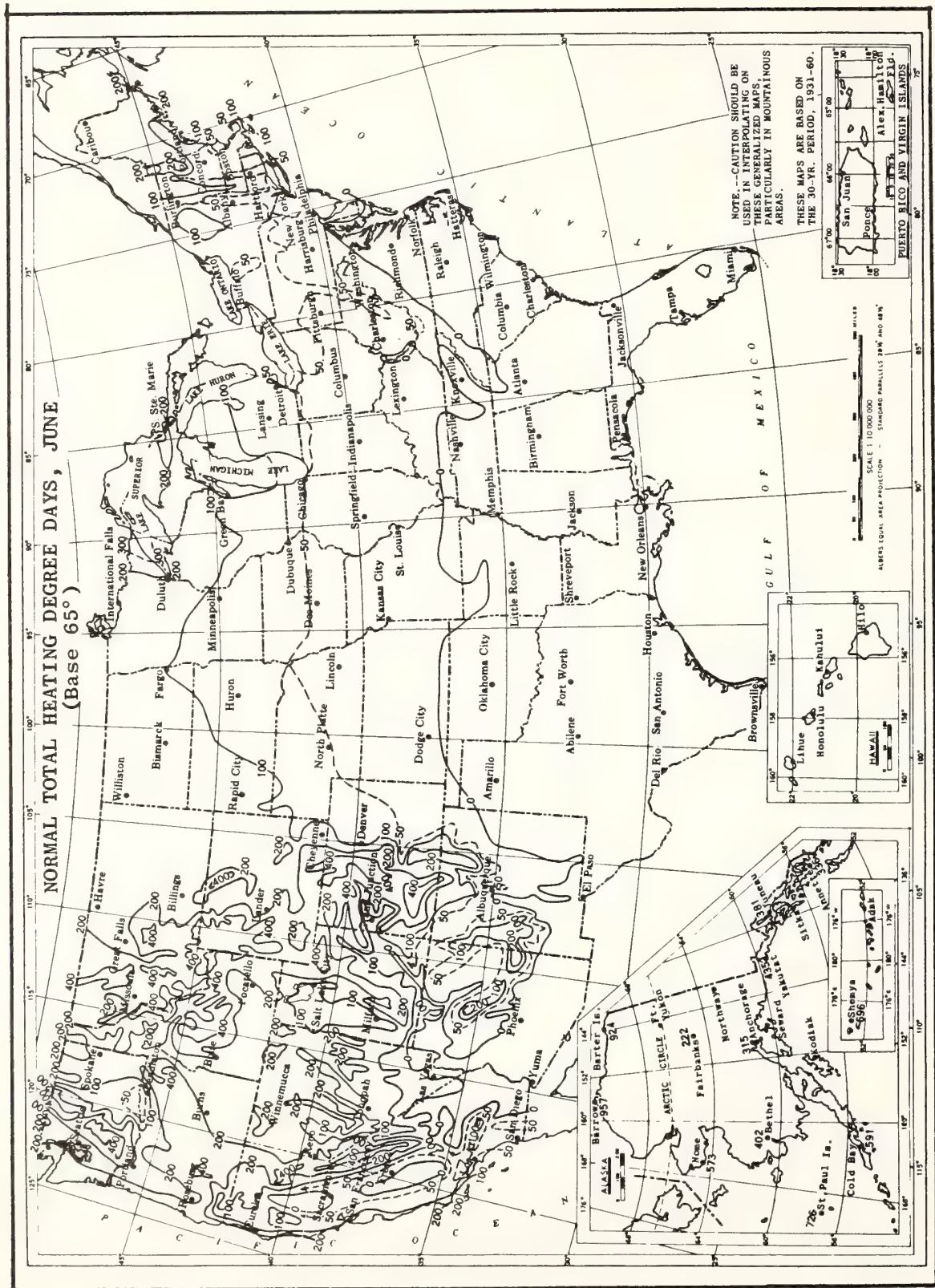


Figure A5-7

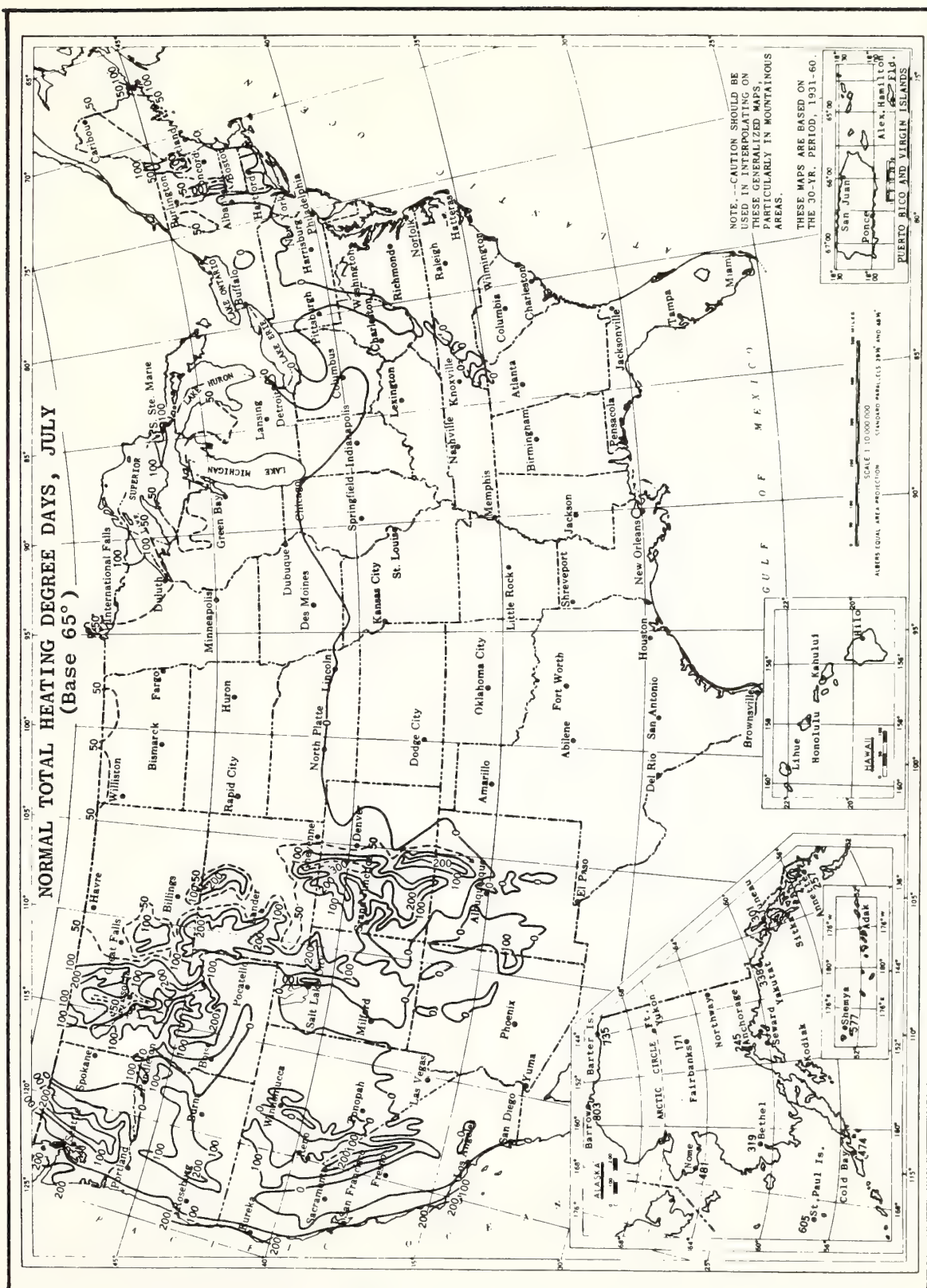


Figure A5-8



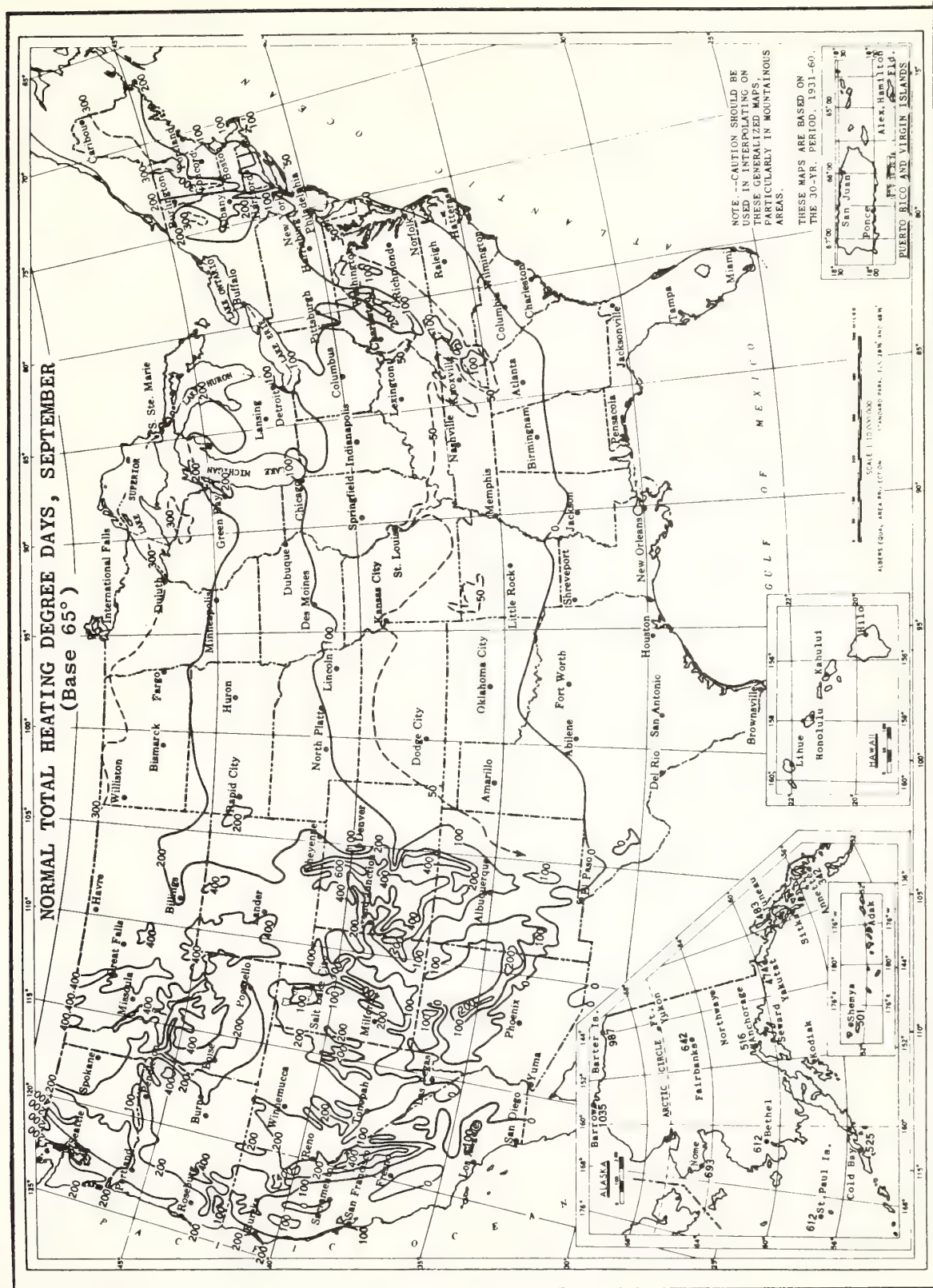
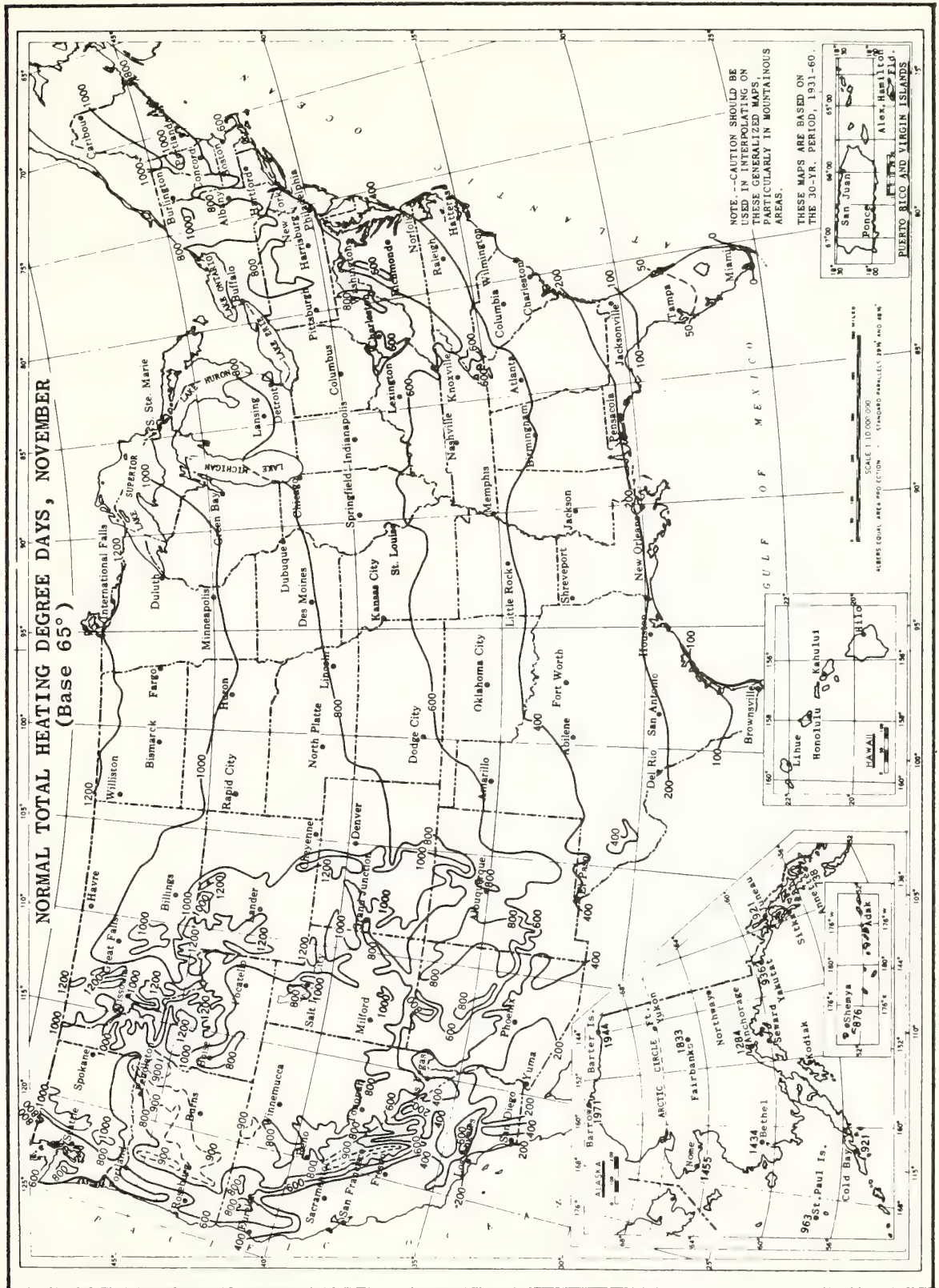
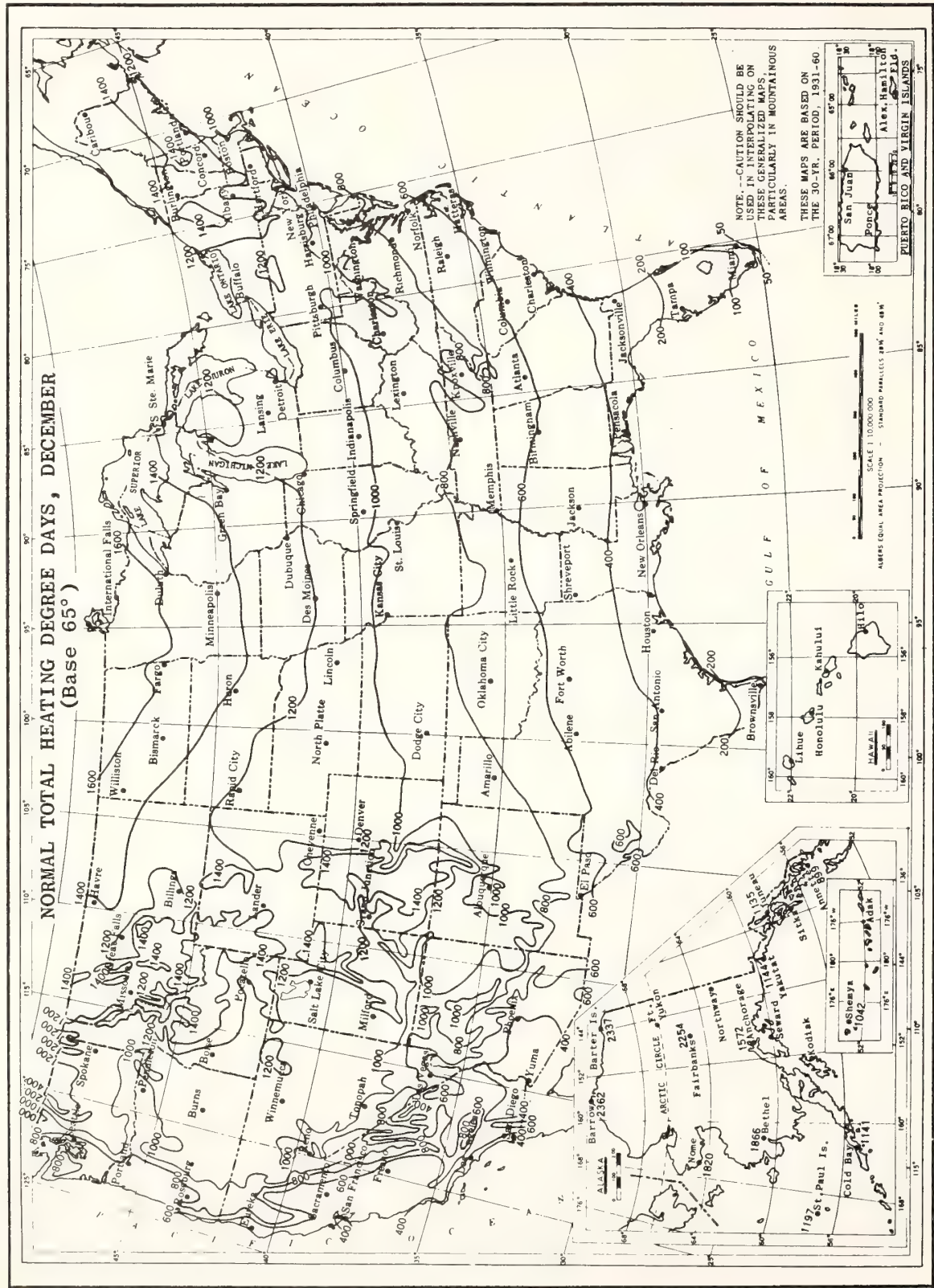


Figure A5-10



Figure A5-11





TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 6

SYSTEM SIZING AND APPROXIMATE METHODS
FOR ESTIMATING SYSTEM PERFORMANCE

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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OBJECTIVE

The primary objective in this module is to present simple methods for estimating the fraction of annual heating load that can be supplied by a specific solar system at a given location.

INTRODUCTION

There have been various methods devised for estimating the quantity of thermal energy that could be supplied by a specific solar system at a particular location during a given year. The methods range from simple tabular or graphical methods to very complex numerical simulation models which require very large computing capability. In this module the simple procedures are discussed and in Module 9 a detailed design method is explained. The very complex methods are used more for research purposes than for design and are not discussed in this manual. However, it may be useful to know that detailed computer methods for analyzing transient behavior of solar systems are available. Information on procurement of the computer programs may be obtained by writing to the Solar Energy Research Institute, Golden, Colorado, or to organizations that developed the computer programs, such as the Solar Energy Laboratory, University of Wisconsin, Madison (TRNSYS) and Los Alamos Scientific Laboratory (DOE2).

SIZING THE SYSTEM

System component sizes are selected on the basis of collector area, and collector area is determined by economic evaluation or is arbitrarily selected to provide a given fraction of the annual heating load for space and domestic water. Rules of thumb are generally used to select component sizes once collector area is determined. Simple methods to estimate system performance assume average or recommended component sizes, while detailed performance estimation methods take into account sizes of the components. Rules of thumb are provided for collector tilt and orientation, fluid flow rate through the collector, storage size and heat exchanger sizes. Standard design procedures are used to select pumps and blowers, as well as pipe and duct sizes. Auxiliary furnaces or boilers are sized according to design heat loss rate calculations in the manner recommended by ASHRAE, SMACNA, NAHB and others; a condensed version is provided in Module 5 of this manual.

The rules of thumb for sizing components of liquid- and air-type solar systems for space and domestic water heating are provided in Tables 6-1 and 6-2. From collector area and unit flow rates the volume flow rate of pumps and blowers can be determined, but pressure drops must be calculated from piping and ducting layouts.

APPROXIMATE METHODS FOR COLLECTOR SIZING

USES OF APPROXIMATE METHODS

It is important for designers of solar space and domestic water heating systems to be able to estimate system performance quickly,

Table 6-1

Rules of Thumb for Sizing Components of Liquid Systems

Space Heating System		
Component	Recommended Value	Range of Values
Collector orientation	South	Southeast to Southwest
Collector tilt	latitude $+15^\circ$	latitude -20° to vertical
Collector fluid flow rate	0.02 gpm/ft^2	$0.01 \text{ to } 0.03 \text{ gpm/ft}^2$
Water storage	2 gal/ft^2	$1.5 \text{ to } 2.5 \text{ gal/ft}^2$
Collector/storage Heat exchanger	$F_R'/F_R = 0.97$	$F_R'/F_R > 0.9$
Load heat exchanger	$\frac{\epsilon_L C_{\min}}{(UA)_L} = 2$	$1 < \frac{\epsilon_L C_{\min}}{(UA)_L} < 3$
Domestic Water Heating System Only		
Component	Recommended Value	Range of Values
Collector orientation	South	Southeast to Southwest
Collector tilt	latitude	latitude -35° to latitude $+35^\circ$
Collector fluid flow rate	0.02 gpm/ft^2	$0.01 \text{ to } 0.03 \text{ gpm/ft}^2$
Preheat storage tank volume	2 times DHW auxiliary tank	1.5 to 2.5 times DHW auxiliary tank
Double-wall heat exchanger	$\epsilon_{HX} = 0.7$ $F_R'/F_R = 0.95$	$0.5 < \epsilon_{HX} < 0.9$ $F_R'/F_R > 0.9$

Table 6-2

Rules of Thumb for Sizing Components of Air Systems

Space Heating System		
Component	Recommended Value	Range of Values
Collector orientation	South	Southeast to Southwest
Collector slope	latitude +15°	latitude -20° to vertical
Collector air flow rate	2 cfm/ft ²	1.5 to 3 cfm/ft ²
Pebble-bed storage	0.75 ft ³ /ft ²	0.5 to 1 ft ³ /ft ²
Pebble size	0.75-1.5 in.	Uniform sizes 0.5 to 3-in.
Pebble-bed depth	6 ft	4 to 8 ft
Pressure drop in pebble bed	0.15 in W.G.	0.1 to 0.3-in W.G.
Pressure drop in ducts	0.08-in W.G./100 ft	0.06 to 0.10-in. W.G./100 ft
Duct insulation	R-7	R-4 to R-13
Domestic water heat exchanger	$\epsilon_{HX} = 0.4$	$0.2 < \epsilon_{HX} < 0.7$
Domestic water preheater tank volume	2 times DHW auxiliary tank	1.5 to 2.5 times DHW auxiliary tank
Domestic Water Heating System Only		
Component	Recommended Value	Range of Values
Collector orientation	South	Southeast to Southwest
Collector storage	latitude	-35° to latitude +35°
Collector air flow rate	2 cfm/ft ²	1.0 to 3 cfm/ft ²
Heat exchanger	$\epsilon_{HX} = 0.7$	$0.5 < \epsilon_{HX} < 0.9$
Preheater tank volume	2 times DHW auxiliary tank	1.5 to 2.5 times DHW auxiliary tank

particularly in addressing the questions of a client. If there is easy access to a computer, detailed answers can be provided quickly using a complex performance estimation model, but because not everyone concerned has such access, rapid estimation procedures using hand-held calculators will be useful. Also, during the planning stages, approximate methods may be used advantageously to examine different system sizes for a particular application.

There are several approximate methods for estimating system performance. Two are included in the module. The accuracies of estimation methods are dependent on location and system type. The numerical values resulting from these calculations may differ from more detailed performance estimation methods by as much as 20 percent.

SYSTEMS TYPES ASSUMED

The basic arrangements assumed for space and water heating systems are shown in Figures 6-1 through 6-4. In general, the calculations apply only to the types of systems shown. The methods are not applicable for swimming pool heaters, solar-assisted heat pumps, nor for systems using concentrating collectors.

GRAPHICAL METHOD

A simple graphical approach for estimating the fraction of annual space heating load that can be supplied by a solar system is shown in Figure 6-5. The curve can also be used to estimate the collector area required to meet a pre-selected solar fraction. The graphical method is illustrated by the examples given below:

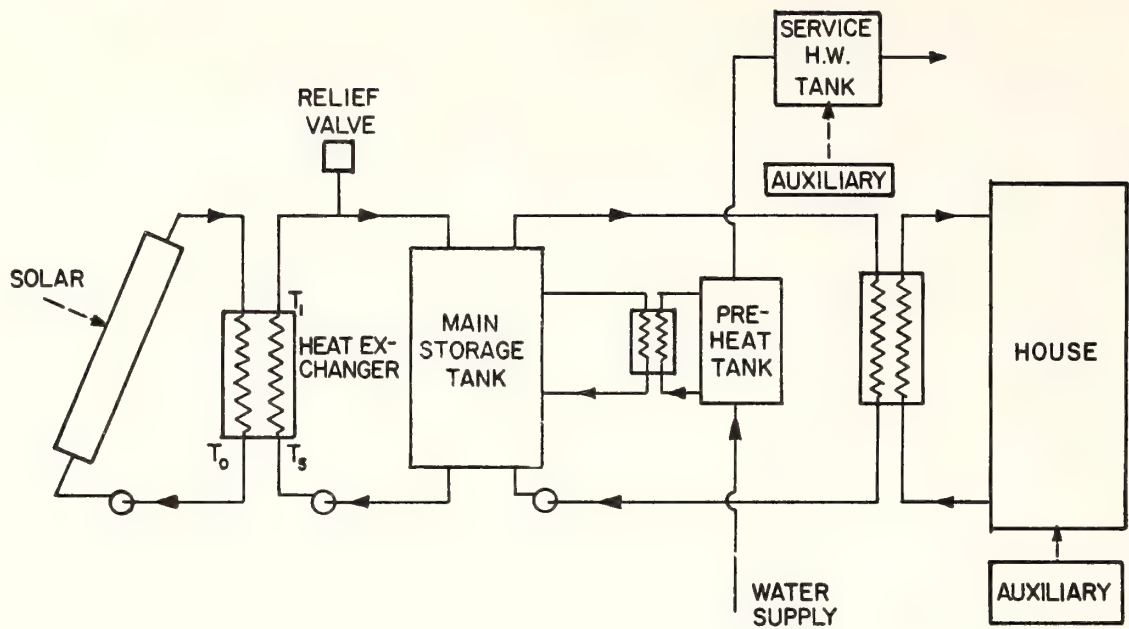


Figure 6-1. Schematic Diagram of a Liquid-Based Solar Space and Water Heating System

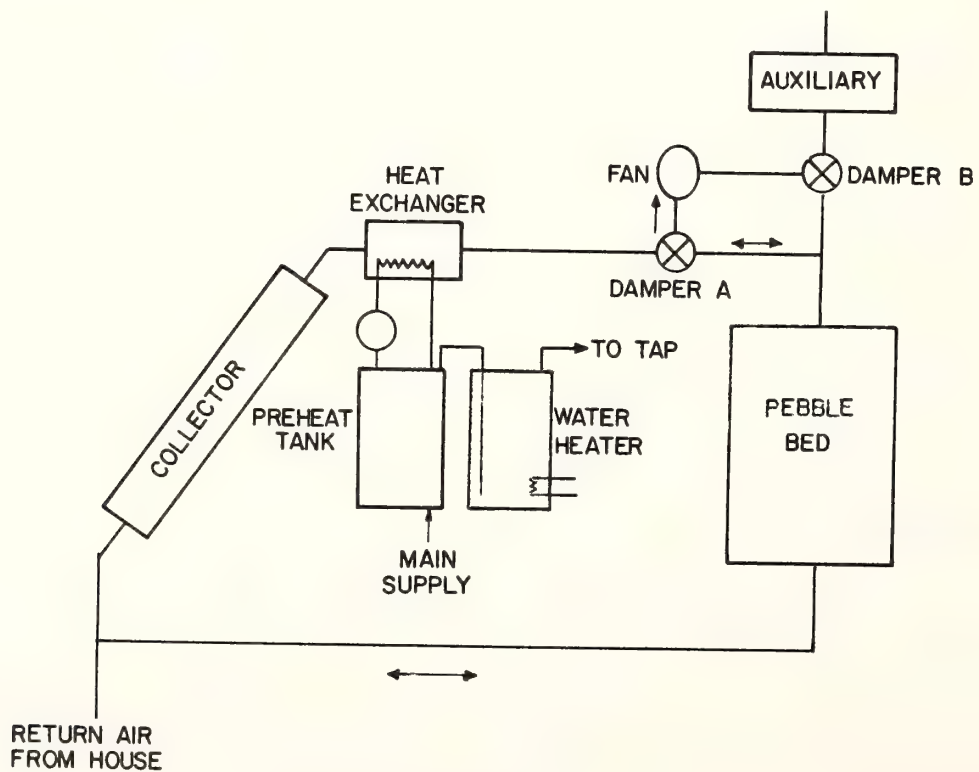


Figure 6-2. Schematic Diagram of an Air-Based Solar Space and Water Heating System

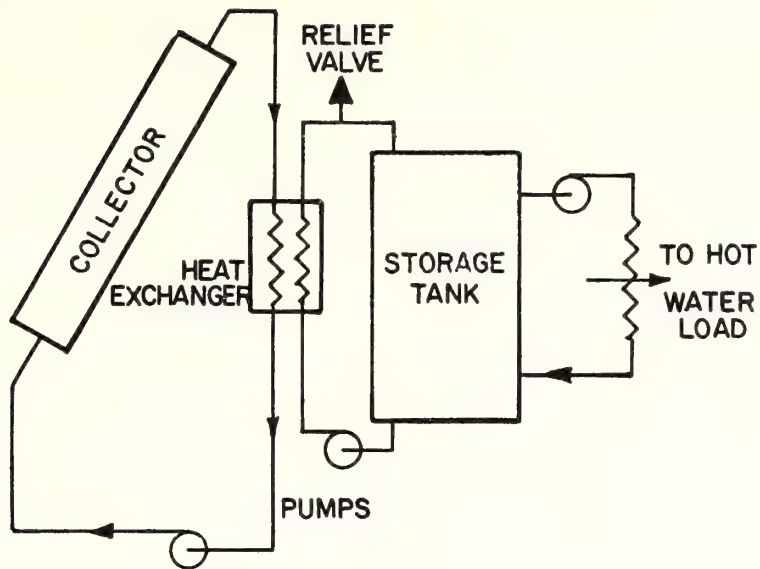


Figure 6-3. Schematic Diagram of a Liquid-Based Domestic Water Heating System

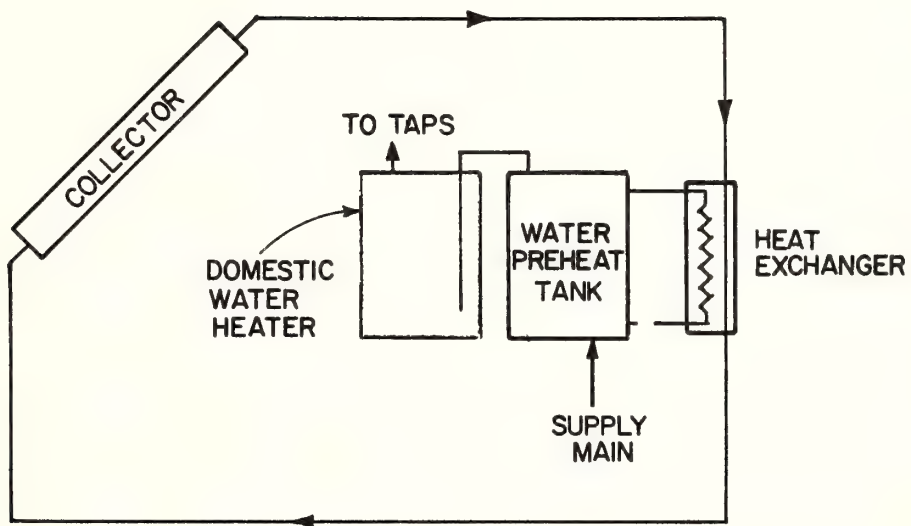


Figure 6-4. Schematic Diagram of an Air-Based Domestic Water Heating System

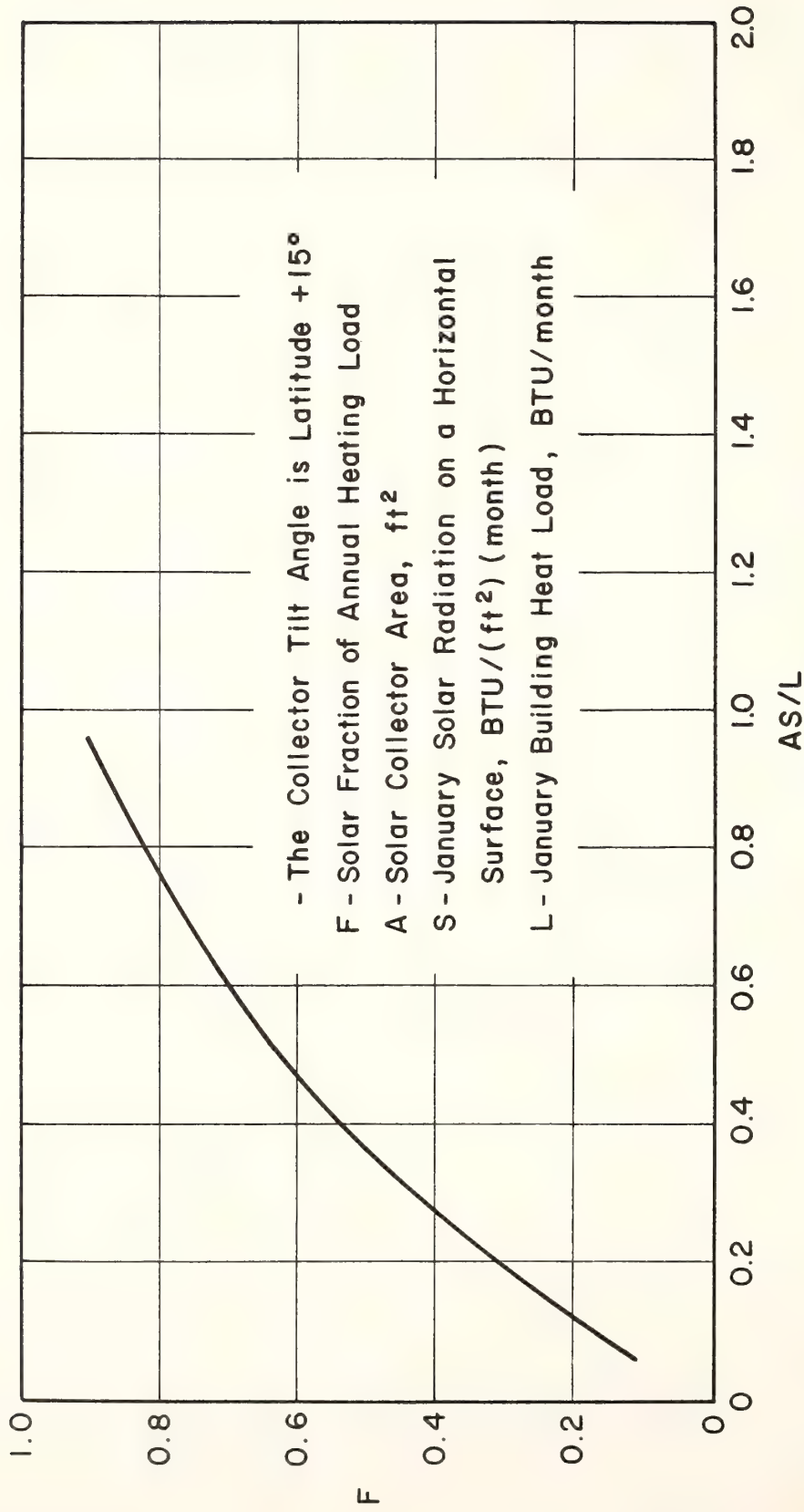


Figure 6-5. Fraction of Annual Heating Load Supplied by Solar Heating Systems

Example 6-1 - Estimate the fraction of annual heating load that could be supplied by a liquid-heating solar system having 300 ft^2 of collectors in a building that has an effective thermal conductance, $(UA)_L$, of $400 \text{ Btu}/(\text{hr} \cdot ^\circ\text{F})$ and is located in Denver, Colorado.

Solution - The parameter SA/L is the ratio of available solar energy in January divided by the building load for January.

From Module 3, $\bar{I} = 742 \text{ Btu}/\text{ft}^2 \cdot \text{day}$

Therefore, $S = \bar{I} \cdot n = 742 \times 31 = 23002 \text{ Btu}/\text{ft}^2 \cdot \text{mo}$

and $SA = 23002 \times 300 = 6.9 \times 10^6 \text{ Btu}/\text{mo}$

From Module 5, $L = (UA)_L \times 24 \times DD \text{ Btu}/\text{mo}$

$$L = 400 \times 1132 \times 24 \text{ Btu}/\text{mo}$$

$$L = 10.87 \times 10^6 \text{ Btu}/\text{mo}$$

$$SA/L = 6.9/10.87 = 0.63$$

From Figure 6-5, $F = 0.72$

Example 6-2 - Determine the area of collectors required to supply 50 percent of the annual heating load for the building of Example 6-1.

Solution - From Figure 6-5, for $F = 0.5$,

$$SA/L \approx 0.37$$

$$A = \frac{0.37 \times L}{S} = \frac{0.37 \times 10.87 \times 10^6}{23000} = 175 \text{ ft}^2$$

The graphical method illustrated above for estimating system performance (in terms of F) is very rapidly completed but a single curve cannot reflect different types of systems, different collectors, nor effects of geographical location, unless individualized curves are developed for every different circumstance.

Other graphical procedures use a more refined approach by calculating monthly solar fractions based on monthly solar-to-load ratios, but

the reliability of the results is only slightly improved. A universal curve of the type shown in Figure 6-5 is not possible with the variety of systems available.

RELATIVE AREAS METHOD

If the parameter of the horizontal axis in Figure 6-5 is expressed as a relative collector area, where the basis is collector area to supply approximately 50 percent of the load, the parameter AS/L could be re-expressed as

$$\frac{SA/L}{SA_0/L} = \frac{A}{A_0}, \quad (6-1)$$

where A_0 is the collector area which will supply approximately 50 percent of the load.

A curve similar to Figure 6-5 could be drawn and the curve could be described by a general equation:

$$F = C_1 + C_2 \ln (A/A_0), \quad (6-2)$$

where C_1 and C_2 are constants for a given system type and location

A is a selected collector area.

The value of A_0 is dependent upon the heating load, and the collector and system characteristics as expressed below:

$$A_0 = \frac{A_s(UA)_L}{F_R' \alpha - F_R' U_L (Z)} \quad (6-3)$$

where $F_R' \alpha$ is a collector characteristic (dimensionless)

$F_R' U_L$ is a collector loss characteristic $\text{Btu}/(\text{hr} \cdot \text{ft}^2 \cdot ^\circ\text{F})$

$(UA)_L$ is the effective thermal conductance of the building Btu/(hr·°F)

A_s is a location-dependent value which reflects the thermal building load (hr·ft²·°F)/Btu

Z is a location- and system-dependent value which affects collection losses (hr·ft²·°F)/Btu.

For any given location and system type, with knowledge of 4 values, C_1 , C_2 , A_s and Z , and the collector characteristics, $F_R'\tau\alpha$ and $F_R'U_L$, the annual solar fraction for the system may be determined. Values of C_1 , C_2 , A_s and Z for a number of locations are listed in Table 6-3 on pages 6-14 to 6-16. The relative areas method is illustrated in Example 6-3.

Example 6-3 - Estimate the fraction of annual space and DHW heating load that could be supplied by a liquid-heating solar system having 300 ft² of Miromit collectors, for a building that has an effective thermal conductance $(UA)_L$ of 400 Btu/(hr·°F) and is located in Denver, Colorado.

Solution -

Collector characteristics (from Module 4)

$$F_R'\tau\alpha = 0.724 \quad F_R'U_L = 0.947 \text{ Btu}/(\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F})$$

From Table 6-3

$$A_s = 0.175 \quad Z = 0.197 \quad C_1 = .538 \quad C_2 = .316$$

Assume a heat exchanger factor, $F_R'/F_R = 0.97$

Using Equation (6-3)

$$A_o = \frac{0.175(400)}{(0.97)(0.724) - (0.97)(0.947)(0.197)} = 134.3 \text{ ft}^2$$

$$F = 0.538 + 0.312 \ln (300/134.3) = \underline{0.79}$$

Comparing the result with Example (6-1) it is noted that the difference is about 10 percent in this particular example, with the relative areas method considered to be more reliable than the single-graph method.

For a domestic water heating system of the type illustrated in Figure 6-3, the calculation of A_o changes slightly to

$$A_o = \frac{A_D D \Delta T \times 10^{-3}}{F_{R\tau\alpha} - F_{R'L}(Z)} \quad (6-4)$$

where

A_D is a location-dependent variable ($\text{ft}^2 \cdot \text{day} / (^\circ\text{F} \cdot \text{kgal})$)

D is daily hot water use (gal/day)

ΔT is average yearly temperature rise from mains temperature to delivery temperature ($^\circ\text{F}$)

Example 6-4 - Determine the number of collector modules required to provide hot water for a family of four in a residential building in Kansas City, Mo., if the solar system is to supply 60 percent of the yearly hot water requirement. Collector characteristics in $F_{R\tau\alpha} = 0.724$, $F_{R'L} = 0.947 \text{ Btu}/(\text{hr} \cdot \text{ft}^2 \cdot ^\circ\text{F})$, and module size is 3 ft by 8 ft.

Solution - Assume an average family of four requires 80 gallons of hot water per day, and average $\Delta T = (140^\circ - 50^\circ) = 90^\circ\text{F}$.

From Table 6-3

$$A_D = 3.357 \quad Z = 0.234 \quad C_1 = 0.541 \quad C_2 = 0.332$$

Assume $F_{R'L}/F_R = 0.95$

Using Equation (6-4)

$$A = \frac{3.357 \times 80 \times 90 \times 10^{-3}}{(0.95)(0.724) - (0.95)(0.947)(0.234)} = 50.6 \text{ ft}^2$$

From Equation (6-2)

$$F = C_1 + C_2 \ln (A/A_0)$$

$$0.6 = 0.341 + 0.332 \ln (A/A_0)$$

$$\ln(A/A_0) = 0.178$$

$$A/A_0 = e^{0.178} = 1.194$$

$$A = 50.6 \times 1.194 = 60.4 \text{ ft}^2$$

Since 1 module = 24 ft², 2.52 modules are required, but fractional modules are not obtainable. Hence 3 collector modules are required, and more than 60 percent of the hot water needs will be met by solar in average years.

In using the relative areas method, it is necessary to calculate the natural logarithm, or the antilogarithm of a number. Most hand-held calculators designed for scientific work include the \ln function and its inverse, so no difficulty is encountered in the calculations. However, for the benefit of trainees without such calculators, an abbreviated table of natural logarithms of numbers is provided in this module.

Table 6-3
Constants for Relative Areas Method

CITY	ST	LAT. (DEG.)	D-H (HR/YR)	LIQUID SYSTEM			AIR SYSTEM			DOMESTIC HOT WATER ONLY					
				A_s ($\frac{^\circ\text{F}\cdot\text{ft}^2\cdot\text{Hr}}{\text{Btu}}$)	Z	C_1	C_2	A_s ($\frac{^\circ\text{F}\cdot\text{ft}^2\cdot\text{Hr}}{\text{Btu}}$)	Z	C_1	C_2	A_D ($\frac{^\circ\text{F}\cdot\text{ft}^2\cdot\text{Hr}}{\text{Btu}}$)	Z	C_1	C_2
Annette	AK	55.0	172607	.309	.390	.510	.257	.289	.316	.514	.284	5.007	.392	.530	.309
Bethel	AK	60.5	316702	.485	.365	.508	.255	.453	.289	.511	.284	4.409	.417	.535	.314
Fairbanks	AK	64.5	342696	.638	.412	.506	.226	.595	.331	.505	.259	4.288	.407	.538	.322
Matanuska	AK	61.3	260371	.483	.413	.503	.231	.450	.330	.502	.253	4.818	.418	.539	.322
Birmingham	AL	33.3	68248	.161	.256	.529	.283	.152	.210	.548	.319	3.370	.211	.545	.341
Fort Smith	AR	35.2	80060	.195	.268	.528	.284	.184	.217	.547	.322	3.412	.217	.543	.338
Little Rock	AR	34.4	77255	.192	.268	.528	.282	.181	.221	.547	.320	3.421	.219	.543	.337
Page	AZ	36.4	129119	.149	.172	.535	.312	.140	.141	.545	.335	2.423	.177	.552	.356
Phoenix	AZ	33.3	37249	.063	.163	.536	.309	.059	.133	.543	.338	2.398	.134	.558	.367
Tucson	AZ	32.1	43199	.070	.159	.540	.314	.066	.130	.551	.336	2.396	.141	.557	.365
Yuma	AZ	32.4	24141	.046	.176	.530	.303	.043	.141	.533	.322	2.499	.132	.559	.369
Davis	CA	38.3	60047	.137	.269	.516	.257	.129	.220	.528	.286	3.066	.220	.540	.331
Fresno	CA	36.5	62665	.130	.243	.519	.262	.123	.199	.531	.288	2.957	.188	.545	.340
Yokern	CA	35.4	56878	.077	.140	.534	.307	.072	.115	.541	.325	2.052	.121	.559	.369
Los Angeles	CA	33.6	43657	.054	.197	.535	.305	.051	.154	.544	.330	2.834	.182	.551	.354
Pasadena	CA	34.1	40647	.060	.201	.535	.308	.057	.159	.546	.334	2.916	.183	.550	.353
Riverside	CA	33.6	46056	.065	.173	.538	.312	.061	.142	.551	.336	2.614	.165	.554	.359
Sacramento	CA	38.3	68225	.133	.255	.518	.259	.125	.206	.526	.286	3.028	.197	.543	.336
San Diego	CA	32.4	36161	.053	.211	.536	.309	.050	.168	.547	.336	3.139	.200	.548	.347
San Francisco	CA	37.5	73911	.082	.227	.530	.310	.077	.180	.542	.340	3.232	.221	.543	.338
San Jose	CA	37.2	55967	.101	.254	.520	.270	.095	.206	.531	.294	3.336	.226	.541	.334
Santa Maria	CA	34.5	71207	.066	.183	.529	.310	.062	.146	.538	.334	2.600	.180	.551	.354
Boulder	CO	40.0	132960	.196	.241	.531	.301	.184	.196	.544	.334	3.295	.242	.540	.333
Denver	CO	39.5	144377	.175	.197	.538	.316	.165	.156	.549	.347	2.639	.197	.548	.348
Grand Junction	CO	39.1	135334	.196	.201	.531	.303	.185	.165	.544	.331	2.661	.193	.549	.349
Grand Lake	CO	40.2	259248	.262	.230	.526	.302	.246	.185	.537	.332	2.876	.256	.537	.327
Pueblo	CO	38.2	129448	.164	.184	.537	.316	.154	.147	.548	.345	2.547	.184	.551	.353
Hartford	CT	41.6	152395	.286	.286	.526	.287	.269	.232	.543	.328	3.653	.273	.536	.323
Washington	DC	38.5	101375	.237	.237	.520	.271	.224	.248	.533	.307	3.829	.260	.537	.325
Apalachicola	FL	29.5	31391	.066	.195	.538	.311	.062	.159	.550	.339	2.921	.168	.553	.358
Gainesville	FL	29.4	25941	.058	.202	.535	.306	.054	.164	.544	.334	3.057	.171	.553	.358
Jacksonville	FL	30.3	31843	.076	.217	.537	.309	.072	.177	.551	.341	3.292	.189	.550	.352
Key West	FL	24.3	1533	.004	.172	.544	.326	.004	.143	.562	.359	2.993	.145	.557	.365
Miami	FL	25.5	5137	.014	.160	.542	.333	.013	.132	.562	.368	2.919	.149	.556	.364
Pensacola	FL	30.3	37866	.093	.230	.534	.299	.087	.188	.548	.331	3.215	.187	.550	.351
Tallahassee	FL	30.3	37510	.076	.192	.535	.305	.072	.156	.545	.331	3.045	.178	.550	.352
Tampa	FL	27.6	17233	.036	.179	.536	.318	.033	.146	.544	.346	2.873	.156	.555	.363
Atlanta	GA	33.4	74282	.162	.240	.529	.289	.153	.198	.547	.342	3.338	.214	.545	.343
Griffin	GA	33.2	67201	.141	.228	.533	.291	.133	.187	.550	.334	3.142	.201	.548	.347
Macon	GA	32.4	53756	.122	.226	.536	.300	.115	.186	.553	.334	3.189	.192	.549	.350
Savannah	GA	32.1	46844	.108	.228	.534	.298	.102	.186	.550	.331	3.265	.193	.549	.351
Hilo	HI	19.4	0	.000	.000	.000	.000	.000	.000	.000	.000	3.470	.185	.550	.352
Honolulu	HI	21.2	0	.000	.000	.000	.000	.000	.000	.000	.000	2.617	.132	.558	.367
Ames	IA	42.0	163776	.312	.284	.525	.281	.295	.231	.542	.327	3.567	.271	.537	.325
Des Moines	IA	41.3	161033	.327	.295	.522	.269	.308	.243	.538	.308	3.495	.262	.537	.324
Boise	ID	43.3	139417	.238	.277	.515	.259	.223	.225	.524	.290	3.183	.229	.536	.323
Pocatello	ID	42.6	169513	.242	.248	.522	.278	.228	.197	.531	.308	2.960	.229	.540	.331
Twin Falls	ID	40.4	151774	.263	.288	.518	.264	.247	.233	.529	.296	3.458	.260	.534	.319
Chicago	IL	41.6	147047	.314	.303	.522	.276	.296	.247	.540	.315	3.716	.270	.535	.321
Lemont	IL	41.4	147047	.316	.305	.522	.275	.298	.250	.540	.321	3.730	.271	.535	.321
Peoria	IL	40.4	146345	.315	.308	.519	.265	.296	.253	.535	.302	3.591	.263	.536	.322
Fort Wayne	IN	41.0	149012	.319	.315	.518	.262	.301	.261	.536	.304	3.651	.270	.534	.318
Indianapolis	IN	39.4	133847	.325	.340	.518	.257	.306	.279	.532	.327	3.857	.276	.532	.314
Dodge City	KS	37.5	121103	.176	.196	.534	.307	.166	.162	.549	.337	2.699	.191	.549	.350
Kansas City	KS	39.1	124369	.242	.269	.524	.282	.227	.218	.538	.316	3.411	.238	.541	.333
Wichita	KY	37.4	112481	.206	.237	.530	.293	.195	.189	.545	.303	3.103	.212	.546	.343
Lexington	KY	38.0	113495	.227	.268	.523	.271	.215	.216	.537	.327	3.162	.221	.540	.331

Table 6-3 (continued)

CITY	ST	LAT. (DEG.)	D-H (FHR/YR)	LIQUID SYSTEM			AIR SYSTEM			DOMESTIC HOT WATER ONLY			
				A_s ($^{\circ}\text{F}\cdot\text{ft}^2\cdot\text{Hr.}$ Btu)	Z	C_1	A_s ($^{\circ}\text{F}\cdot\text{ft}^2\cdot\text{Hr.}$ Btu)	Z	C_1	A_D $\text{ft}^2\cdot\text{D.}$ $^{\circ}\text{F}\cdot\text{kgal}(\frac{\text{Btu}}{\text{Btu}})$	Z	C_1	C_2
Louisville	KY	38.1	111353	.266	.311	.521	.252	.253	.539	3.690	.257	.536	.323
Lake Charles	LA	30.1	35015	.087	.231	.534	.082	.189	.549	3.228	.187	.549	.350
New Orleans	LA	29.6	33238	.098	.276	.527	.093	.224	.544	3.896	.225	.542	.336
Shreveport	LA	32.3	52415	.136	.255	.532	.128	.209	.550	3.407	.204	.546	.343
Amherst	MA	42.2	157825	.379	.370	.516	.357	.302	.528	4.277	.323	.528	.308
Blue Hill	MA	42.1	152809	.310	.320	.523	.293	.257	.539	3.956	.299	.532	.316
Boston	MA	42.2	135215	.321	.342	.521	.304	.278	.538	4.198	.302	.531	.314
Lynn	MA	42.3	135215	.341	.370	.516	.322	.303	.528	4.277	.310	.527	.304
Natick	MA	42.2	147455	.303	.307	.522	.285	.248	.537	3.890	.285	.533	.317
Annapolis	MD	38.6	109146	.239	.292	.522	.225	.239	.538	3.675	.256	.538	.327
Baltimore	MD	39.1	113491	.246	.287	.523	.232	.235	.539	3.643	.252	.539	.329
Silver Hill	MD	38.5	101064	.227	.280	.524	.215	.225	.542	3.596	.244	.540	.332
Caribou	ME	46.5	234406	.385	.309	.521	.362	.249	.533	3.805	.320	.529	.309
Portland	ME	43.4	175462	.285	.281	.525	.268	.226	.539	3.18	.275	.535	.321
Detroit	MI	42.1	154049	.361	.351	.517	.340	.291	.532	3.928	.294	.529	.309
East Lansing	MI	42.4	165697	.385	.365	.517	.363	.298	.529	4.071	.313	.527	.305
Lansing	MI	42.5	165689	.368	.347	.519	.348	.284	.533	3.954	.301	.529	.309
Sault Ste. Marie	MI	46.3	217151	.363	.319	.517	.343	.256	.528	3.763	.317	.527	.304
Columbia	MO	38.6	121103	.245	.275	.524	.230	.225	.539	3.359	.236	.540	.331
Kansas City	MO	39.2	123859	.257	.274	.525	.243	.220	.541	3.357	.234	.541	.332
St. Louis	MO	38.5	113993	.253	.285	.522	.240	.232	.540	3.505	.240	.539	.330
Springfield	MO	37.1	109672	.175	.249	.526	.167	.207	.536	2.951	.215	.546	.302
Duluth	MN	46.5	234136	.397	.323	.518	.374	.260	.529	3.749	.315	.530	.312
Minn.-St. Paul	MN	44.5	195807	.398	.325	.519	.376	.265	.535	3.765	.298	.532	.316
St. Cloud	MN	45.3	212831	.349	.274	.525	.328	.223	.539	3.415	.277	.535	.321
Jackson	MS	32.2	55195	.139	.257	.530	.131	.210	.547	3.322	.204	.546	.343
Billings	MT	45.5	174353	.259	.256	.527	.245	.204	.540	3.139	.245	.539	.329
Glasgow	MT	48.1	215906	.294	.237	.527	.277	.192	.539	2.952	.242	.540	.332
Great Falls	MT	47.3	186001	.268	.259	.524	.283	.251	.536	3.202	.252	.537	.326
Summit	MT	48.2	255070	.393	.349	.513	.367	.282	.521	4.009	.348	.524	.299
Asheville	NC	35.3	101681	.177	.243	.527	.166	.196	.539	3.203	.221	.544	.340
Cape Hatteras	NC	35.2	65545	.117	.208	.531	.111	.167	.543	2.863	.183	.551	.354
Charlotte	NC	35.1	77225	.159	.229	.532	.150	.190	.550	3.178	.205	.547	.346
Greensboro	NC	36.1	91801	.189	.252	.525	.179	.207	.544	3.329	.229	.543	.338
Greenville-Spartanburg	NC	34.5	75907	.162	.237	.531	.153	.195	.548	3.243	.209	.546	.344
Raleigh	NC	35.5	84335	.161	.231	.530	.152	.184	.545	3.306	.216	.546	.343
Raleigh-Durham	NC	35.5	84333	.173	.241	.526	.164	.198	.545	3.388	.221	.545	.341
Bismarck	ND	46.5	217056	.327	.260	.524	.307	.211	.537	3.149	.260	.538	.327
Fargo	ND	46.5	222497	.415	.319	.519	.391	.257	.530	3.657	.300	.533	.317
Lincoln	NE	40.5	140406	.256	.257	.527	.242	.212	.545	3.328	.242	.540	.332
North Omaha	NE	41.2	158687	.263	.249	.528	.249	.199	.543	3.321	.237	.542	.335
Atlantic City	NJ	39.3	112629	.218	.261	.525	.206	.213	.541	3.449	.242	.540	.332
Trenton	NJ	40.1	118727	.243	.276	.523	.229	.227	.541	3.562	.253	.539	.329
Albuquerque	NM	35.0	103006	.133	.167	.538	.125	.136	.549	2.332	.164	.554	.360
Ely	NV	39.2	185590	.197	.191	.534	.186	.156	.546	2.536	.206	.547	.345
Las Vegas	NV	36.1	65015	.102	.163	.539	.096	.133	.550	2.377	.144	.557	.365
Reno	NV	39.3	144521	.158	.189	.533	.149	.154	.548	2.511	.192	.548	.348
Albany	NY	42.4	165311	.328	.322	.512	.307	.255	.519	3.829	.292	.531	.313
Binghamton	NY	42.1	174835	.371	.351	.517	.351	.286	.530	3.987	.309	.529	.309
Ithaca	NY	42.3	169246	.407	.387	.518	.385	.319	.529	4.281	.332	.525	.301
New York	NY	40.5	115462	.326	.369	.520	.306	.304	.533	4.503	.310	.530	.311
Rochester	NY	43.1	161249	.348	.338	.518	.329	.275	.532	3.853	.292	.530	.311
Schenectady	NY	42.5	163610	.409	.385	.521	.386	.317	.531	4.646	.344	.530	.311
Syracuse	NY	43.1	160264	.386	.372	.519	.364	.304	.530	4.126	.313	.527	.304
Cleveland	OH	41.2	147697	.367	.372	.516	.346	.307	.526	4.059	.301	.526	.303
Columbus	OH	40.0	135840	.326	.348	.515	.307	.286	.530	3.903	.284	.530	.310
Dayton	OH	39.5	135377	.294	.306	.520	.277	.252	.536	3.580	.259	.535	.321

Table 6-3 (continued)

CITY	ST	LAT. (DEG.)	D-H (FHR/YR)	LIQUID SYSTEM			AIR SYSTEM			DOMESTIC HOT WATER ONLY				
				A_s ($\frac{^{\circ}\text{F}\cdot\text{ft}^2\cdot\text{Hr}}{\text{Btu}}$)	Z	C_1	C_2	A_s ($\frac{^{\circ}\text{F}\cdot\text{ft}^2\cdot\text{Hr}}{\text{Btu}}$)	Z	C_1	C_2	A_D ($\frac{^{\circ}\text{F}\cdot\text{ft}^2\cdot\text{Hr}}{\text{Kgal}}$)	Z	C_1
Put-in-Bay	OH	41.4	142030	.376	.369	.519	.254	.354	.303	.532	3.946	.288	.526	.303
Oklaoma City	OK	35.2	88676	.159	.206	.535	.304	.151	.169	.552	2.885	.190	.549	.349
Stillwater	OK	36.1	87143	.180	.237	.529	.290	.170	.194	.546	3.173	.207	.545	.342
Tulsa	OK	36.1	88315	.196	.250	.529	.291	.186	.204	.548	3.320	.214	.544	.340
Astoria	OR	46.1	127078	.222	.359	.509	.242	.208	.291	.518	4.422	.333	.521	.293
Corvallis	OR	44.3	116497	.242	.378	.508	.224	.220	.302	.513	4.033	.301	.518	.286
Medford	OR	42.2	118319	.235	.331	.505	.229	.220	.264	.512	3.500	.256	.525	.300
Portland	OR	45.4	115001	.269	.401	.517	.243	.253	.324	.521	4.645	.334	.522	.295
Philadelphia	PA	39.5	116755	.247	.283	.523	.279	.232	.232	.539	3.610	.253	.539	.329
Pittsburgh	PA	40.3	126665	.278	.306	.519	.268	.262	.252	.536	3.700	.262	.535	.322
State College	PA	40.5	147169	.352	.357	.518	.257	.332	.293	.532	4.095	.304	.529	.310
Newport	RI	41.3	139294	.258	.295	.524	.281	.243	.239	.540	3.741	.275	.535	.321
Charleston	SC	32.5	48792	.112	.221	.537	.305	.105	.181	.554	3.243	.195	.549	.350
Rapid City	SD	44.1	176279	.233	.225	.532	.300	.218	.182	.543	3.917	.225	.543	.338
Chattanooga	TN	35.0	84117	.195	.267	.526	.279	.185	.221	.547	3.474	.226	.542	.336
Memphis	TN	35.0	77443	.188	.265	.528	.280	.179	.218	.546	3.342	.211	.544	.338
Nashville	TN	36.1	88703	.232	.301	.522	.266	.219	.248	.542	3.626	.239	.538	.327
Oak Ridge	TN	36.0	94658	.233	.300	.520	.265	.221	.249	.541	3.709	.248	.537	.326
Abilene	TX	32.3	62634	.120	.198	.537	.307	.113	.163	.553	2.867	.174	.552	.356
Amarillo	TX	35.1	100385	.143	.183	.536	.313	.134	.150	.549	3.41	.176	.552	.356
Big Spring	TX	32.2	62182	.121	.200	.537	.305	.114	.163	.552	2.899	.179	.550	.352
Brownsville	TX	25.6	15600	.046	.219	.533	.307	.043	.179	.548	3.182	.168	.552	.355
Corpus Christi	TX	27.5	22318	.061	.240	.527	.301	.057	.191	.535	3.147	.170	.551	.353
Dallas	TX	32.5	54952	.132	.235	.534	.298	.124	.193	.549	3.219	.189	.548	.348
El Paso	TX	31.5	64798	.097	.155	.541	.316	.092	.127	.551	2.330	.145	.557	.366
Fort Worth	TX	32.5	57720	.119	.209	.535	.303	.112	.172	.549	3.397	.225	.544	.339
Houston	TX	29.6	34412	.092	.245	.530	.295	.086	.198	.542	3.260	.183	.549	.349
Midland	TX	31.6	62182	.115	.190	.537	.308	.108	.156	.551	2.753	.172	.553	.357
Port Arthur	TX	29.6	36428	.099	.255	.530	.291	.093	.207	.545	3.447	.196	.547	.346
San Antonio	TX	29.3	37103	.084	.211	.538	.309	.079	.173	.552	3.042	.174	.551	.353
Salt Lake City	UT	40.5	143591	.246	.271	.519	.269	.231	.218	.530	3.244	.237	.538	.328
Mt. Weather	VA	39.0	136034	.257	.280	.525	.285	.242	.226	.541	3.636	.266	.537	.326
Norfolk	VA	36.5	83707	.175	.254	.527	.289	.166	.205	.544	3.397	.225	.544	.339
Richmond	VA	37.5	94530	.206	.267	.523	.280	.195	.219	.542	3.502	.234	.542	.335
Prosser	WA	46.2	134578	.235	.291	.510	.240	.219	.232	.515	3.149	.237	.532	.316
Pullman	WA	46.4	158974	.261	.303	.516	.252	.245	.244	.524	3.398	.266	.529	.310
Richland	WA	46.2	117406	.268	.349	.504	.224	.251	.281	.511	3.644	.262	.526	.303
Seattle	WA	47.3	106178	.251	.413	.509	.221	.235	.335	.510	4.551	.333	.517	.285
Spokane	WA	47.4	164039	.295	.310	.513	.249	.277	.251	.526	3.505	.272	.527	.305
Green Bay	WI	44.3	194344	.387	.336	.519	.260	.364	.272	.530	3.882	.308	.531	.313
Madison	WI	43.1	185520	.348	.310	.522	.270	.329	.248	.534	3.658	.286	.533	.318
Milwaukee	WI	42.6	178649	.350	.325	.518	.261	.331	.262	.532	3.723	.290	.532	.315
Parkersburg	WV	39.2	115601	.301	.352	.519	.258	.286	.285	.533	3.996	.278	.531	.313
Lander	WY	42.5	188879	.211	.193	.536	.313	.198	.157	.547	3.446	.205	.547	.346
Laramie	WY	41.2	212137	.243	.221	.534	.309	.230	.177	.549	2.907	.240	.541	.334
Edmonton	AT	53.3	246430	.376	.300	.517	.267	.355	.238	.528	3.632	.313	.531	.313
Lethbridge	AT	49.4	207455	.300	.274	.520	.273	.283	.221	.532	3.316	.275	.534	.319
Vancouver	BC	48.6	132357	.354	.477	.533	.257	.333	.404	.515	4.03	.403	.526	.299
Churchill	MA	58.5	401471	.542	.322	.517	.276	.508	.262	.530	3.735	.383	.530	.308
Winnipeg	MA	49.5	256296	.405	.286	.522	.276	.382	.231	.534	3.446	.301	.532	.315
Moncton	NB	46.1	209447	.431	.385	.516	.251	.407	.310	.522	4.484	.366	.530	.312
St. Johns	NF	47.3	215785	.472	.442	.525	.268	.445	.365	.520	5.414	.434	.548	.341
Kapuskasing	OT	49.3	277727	.538	.376	.517	.262	.504	.310	.526	4.410	.389	.535	.320
Ottawa	OT	45.3	209638	.391	.316	.522	.269	.368	.258	.536	3.747	.310	.530	.312
Toronto	OT	43.4	163844	.391	.382	.518	.256	.368	.312	.526	4.321	.336	.526	.304
Montreal	QU	45.3	196871	.471	.398	.514	.244	.440	.326	.516	4.368	.354	.526	.302

TABLE OF NATURAL LOGARITHMS

N	0	1	2	3	4	5	6	7	8	9
0.30	-1.204	-1.201	-1.197	-1.194	-1.191	-1.187	-1.184	-1.181	-1.178	-1.174
0.31	-1.171	-1.168	-1.165	-1.162	-1.158	-1.155	-1.152	-1.149	-1.146	-1.143
0.32	-1.139	-1.136	-1.133	-1.130	-1.127	-1.124	-1.121	-1.118	-1.115	-1.112
0.33	-1.109	-1.106	-1.103	-1.100	-1.097	-1.094	-1.091	-1.088	-1.085	-1.082
0.34	-1.079	-1.076	-1.073	-1.070	-1.067	-1.064	-1.061	-1.058	-1.056	-1.053
0.35	-1.050	-1.047	-1.044	-1.041	-1.038	-1.036	-1.033	-1.030	-1.027	-1.024
0.36	-1.022	-1.019	-1.016	-1.013	-1.011	-1.008	-1.005	-1.002	-1.000	-0.997
0.37	-0.994	-0.992	-0.989	-0.986	-0.983	-0.981	-0.978	-0.976	-0.973	-0.970
0.38	-0.968	-0.965	-0.962	-0.960	-0.957	-0.955	-0.952	-0.949	-0.947	-0.944
0.39	-0.942	-0.939	-0.936	-0.934	-0.931	-0.929	-0.926	-0.924	-0.921	-0.919
0.40	-0.916	-0.914	-0.911	-0.909	-0.906	-0.904	-0.901	-0.899	-0.896	-0.894
0.41	-0.892	-0.889	-0.887	-0.884	-0.882	-0.879	-0.877	-0.875	-0.872	-0.870
0.42	-0.868	-0.865	-0.863	-0.860	-0.858	-0.856	-0.853	-0.851	-0.849	-0.846
0.43	-0.844	-0.842	-0.839	-0.837	-0.835	-0.832	-0.830	-0.828	-0.826	-0.823
0.44	-0.821	-0.819	-0.816	-0.814	-0.812	-0.810	-0.807	-0.805	-0.803	-0.801
0.45	-0.799	-0.796	-0.794	-0.792	-0.790	-0.787	-0.785	-0.783	-0.781	-0.779
0.46	-0.777	-0.774	-0.772	-0.770	-0.768	-0.766	-0.764	-0.761	-0.759	-0.757
0.47	-0.755	-0.753	-0.751	-0.749	-0.747	-0.744	-0.742	-0.740	-0.738	-0.736
0.48	-0.734	-0.732	-0.730	-0.728	-0.726	-0.724	-0.722	-0.719	-0.717	-0.715
0.49	-0.713	-0.711	-0.709	-0.707	-0.705	-0.703	-0.701	-0.699	-0.697	-0.695
0.50	-0.693	-0.691	-0.689	-0.687	-0.685	-0.683	-0.681	-0.679	-0.677	-0.675
0.51	-0.673	-0.671	-0.669	-0.667	-0.666	-0.664	-0.662	-0.660	-0.658	-0.656
0.52	-0.654	-0.652	-0.650	-0.648	-0.646	-0.644	-0.642	-0.641	-0.639	-0.637
0.53	-0.635	-0.633	-0.631	-0.629	-0.627	-0.625	-0.624	-0.622	-0.620	-0.618
0.54	-0.616	-0.614	-0.612	-0.611	-0.609	-0.607	-0.605	-0.603	-0.601	-0.600
0.55	-0.598	-0.596	-0.594	-0.592	-0.591	-0.589	-0.587	-0.585	-0.583	-0.582
0.56	-0.580	-0.578	-0.576	-0.574	-0.573	-0.571	-0.569	-0.567	-0.566	-0.564
0.57	-0.562	-0.560	-0.559	-0.557	-0.555	-0.553	-0.552	-0.550	-0.548	-0.546
0.58	-0.545	-0.543	-0.541	-0.540	-0.538	-0.536	-0.534	-0.533	-0.531	-0.529
0.59	-0.528	-0.526	-0.524	-0.523	-0.521	-0.519	-0.518	-0.516	-0.514	-0.512
0.60	-0.511	-0.509	-0.507	-0.506	-0.504	-0.503	-0.501	-0.499	-0.498	-0.496
0.61	-0.494	-0.493	-0.491	-0.489	-0.488	-0.486	-0.485	-0.483	-0.481	-0.480
0.62	-0.478	-0.476	-0.475	-0.473	-0.472	-0.470	-0.468	-0.467	-0.465	-0.464
0.63	-0.462	-0.460	-0.459	-0.457	-0.456	-0.454	-0.453	-0.451	-0.449	-0.448
0.64	-0.446	-0.445	-0.443	-0.442	-0.440	-0.439	-0.437	-0.435	-0.434	-0.432
0.65	-0.431	-0.429	-0.428	-0.426	-0.425	-0.423	-0.422	-0.420	-0.419	-0.417
0.66	-0.416	-0.414	-0.412	-0.411	-0.409	-0.408	-0.406	-0.405	-0.403	-0.402
0.67	-0.400	-0.399	-0.397	-0.396	-0.395	-0.393	-0.392	-0.390	-0.389	-0.387
0.68	-0.386	-0.384	-0.383	-0.381	-0.380	-0.378	-0.377	-0.375	-0.374	-0.373
0.69	-0.371	-0.370	-0.368	-0.367	-0.365	-0.364	-0.362	-0.361	-0.360	-0.358

TABLE OF NATURAL LOGARITHMS

N	0	1	2	3	4	5	6	7	8	9
0.70	-0.357	-0.355	-0.354	-0.352	-0.351	-0.350	-0.348	-0.347	-0.345	-0.344
0.71	-0.342	-0.341	-0.340	-0.338	-0.337	-0.335	-0.334	-0.333	-0.331	-0.330
0.72	-0.329	-0.327	-0.326	-0.324	-0.323	-0.322	-0.320	-0.319	-0.317	-0.316
0.73	-0.315	-0.313	-0.312	-0.311	-0.309	-0.308	-0.307	-0.305	-0.304	-0.302
0.74	-0.301	-0.300	-0.298	-0.297	-0.296	-0.294	-0.293	-0.292	-0.290	-0.289
0.75	-0.288	-0.286	-0.285	-0.284	-0.282	-0.281	-0.280	-0.278	-0.277	-0.276
0.76	-0.274	-0.273	-0.272	-0.270	-0.269	-0.268	-0.267	-0.265	-0.264	-0.263
0.77	-0.261	-0.260	-0.259	-0.257	-0.256	-0.255	-0.254	-0.252	-0.251	-0.250
0.78	-0.248	-0.247	-0.246	-0.245	-0.243	-0.242	-0.241	-0.240	-0.238	-0.237
0.79	-0.236	-0.234	-0.233	-0.232	-0.231	-0.229	-0.228	-0.227	-0.226	-0.224
0.80	-0.223	-0.222	-0.221	-0.219	-0.218	-0.217	-0.216	-0.214	-0.213	-0.212
0.81	-0.211	-0.209	-0.208	-0.207	-0.206	-0.205	-0.203	-0.202	-0.201	-0.200
0.82	-0.198	-0.197	-0.196	-0.195	-0.194	-0.192	-0.191	-0.190	-0.189	-0.188
0.83	-0.186	-0.185	-0.184	-0.183	-0.182	-0.180	-0.179	-0.178	-0.177	-0.176
0.84	-0.174	-0.173	-0.172	-0.171	-0.170	-0.168	-0.167	-0.166	-0.165	-0.164
0.85	-0.163	-0.161	-0.160	-0.159	-0.158	-0.157	-0.155	-0.154	-0.153	-0.152
0.86	-0.151	-0.150	-0.149	-0.147	-0.146	-0.145	-0.144	-0.143	-0.142	-0.140
0.87	-0.139	-0.138	-0.137	-0.136	-0.135	-0.134	-0.132	-0.131	-0.130	-0.129
0.88	-0.128	-0.127	-0.126	-0.124	-0.123	-0.122	-0.121	-0.120	-0.119	-0.118
0.89	-0.117	-0.115	-0.114	-0.113	-0.112	-0.111	-0.110	-0.109	-0.108	-0.106
0.90	-0.105	-0.104	-0.103	-0.102	-0.101	-0.100	-0.099	-0.098	-0.097	-0.095
0.91	-0.094	-0.093	-0.092	-0.091	-0.090	-0.089	-0.088	-0.087	-0.086	-0.084
0.92	-0.083	-0.082	-0.081	-0.080	-0.079	-0.078	-0.077	-0.076	-0.075	-0.074
0.93	-0.073	-0.071	-0.070	-0.069	-0.068	-0.067	-0.066	-0.065	-0.064	-0.063
0.94	-0.062	-0.061	-0.060	-0.059	-0.058	-0.057	-0.056	-0.054	-0.053	-0.052
0.95	-0.051	-0.050	-0.049	-0.048	-0.047	-0.046	-0.045	-0.044	-0.043	-0.042
0.96	-0.041	-0.040	-0.039	-0.038	-0.037	-0.036	-0.035	-0.034	-0.033	-0.031
0.97	-0.030	-0.029	-0.028	-0.027	-0.026	-0.025	-0.024	-0.023	-0.022	-0.021
0.98	-0.020	-0.019	-0.018	-0.017	-0.016	-0.015	-0.014	-0.013	-0.012	-0.011
0.99	-0.010	-0.009	-0.008	-0.007	-0.006	-0.005	-0.004	-0.003	-0.002	-0.001
1.00	0.000	0.001	0.002	0.003	0.004	0.005	0.006	0.007	0.008	0.009
1.01	0.010	0.011	0.012	0.013	0.014	0.015	0.016	0.017	0.018	0.019
1.02	0.020	0.021	0.022	0.023	0.024	0.025	0.026	0.027	0.028	0.029
1.03	0.030	0.031	0.031	0.032	0.033	0.034	0.035	0.036	0.037	0.038
1.04	0.039	0.040	0.041	0.042	0.043	0.044	0.045	0.046	0.047	0.048
1.05	0.049	0.050	0.051	0.052	0.053	0.054	0.054	0.055	0.056	0.057
1.06	0.058	0.059	0.060	0.061	0.062	0.063	0.064	0.065	0.066	0.067
1.07	0.068	0.069	0.070	0.070	0.071	0.072	0.073	0.074	0.075	0.076
1.08	0.077	0.078	0.079	0.080	0.081	0.082	0.083	0.083	0.084	0.085
1.09	0.086	0.087	0.088	0.089	0.090	0.091	0.092	0.093	0.093	0.094

TABLE OF NATURAL LOGARITHMS

N	0	1	2	3	4	5	6	7	8	9
1.10	0.095	0.096	0.097	0.098	0.099	0.100	0.101	0.102	0.103	0.103
1.11	0.104	0.105	0.106	0.107	0.108	0.109	0.110	0.111	0.112	0.112
1.12	0.113	0.114	0.115	0.116	0.117	0.118	0.119	0.120	0.120	0.121
1.13	0.122	0.123	0.124	0.125	0.126	0.127	0.128	0.128	0.129	0.130
1.14	0.131	0.132	0.133	0.134	0.135	0.135	0.136	0.137	0.138	0.139
1.15	0.140	0.141	0.141	0.142	0.143	0.144	0.145	0.146	0.147	0.148
1.16	0.148	0.149	0.150	0.151	0.152	0.153	0.154	0.154	0.155	0.156
1.17	0.157	0.158	0.159	0.160	0.160	0.161	0.162	0.163	0.164	0.165
1.18	0.166	0.166	0.167	0.168	0.169	0.170	0.171	0.171	0.172	0.173
1.19	0.174	0.175	0.176	0.176	0.177	0.178	0.179	0.180	0.181	0.181
1.20	0.182	0.183	0.184	0.185	0.186	0.186	0.187	0.188	0.189	0.190
1.21	0.191	0.191	0.192	0.193	0.194	0.195	0.196	0.196	0.197	0.198
1.22	0.199	0.200	0.200	0.201	0.202	0.203	0.204	0.205	0.205	0.206
1.23	0.207	0.208	0.209	0.209	0.210	0.211	0.212	0.213	0.213	0.214
1.24	0.215	0.216	0.217	0.218	0.218	0.219	0.220	0.221	0.222	0.222
1.25	0.223	0.224	0.225	0.226	0.226	0.227	0.228	0.229	0.230	0.230
1.26	0.231	0.232	0.233	0.233	0.234	0.235	0.236	0.237	0.237	0.238
1.27	0.239	0.240	0.241	0.241	0.242	0.243	0.244	0.245	0.245	0.246
1.28	0.247	0.248	0.248	0.249	0.250	0.251	0.252	0.252	0.253	0.254
1.29	0.255	0.255	0.256	0.257	0.258	0.259	0.259	0.260	0.261	0.262
1.30	0.262	0.263	0.264	0.265	0.265	0.266	0.267	0.268	0.268	0.269
1.31	0.270	0.271	0.272	0.272	0.273	0.274	0.275	0.275	0.276	0.277
1.32	0.278	0.278	0.279	0.280	0.281	0.281	0.282	0.283	0.284	0.284
1.33	0.285	0.286	0.287	0.287	0.288	0.289	0.290	0.290	0.291	0.292
1.34	0.293	0.293	0.294	0.295	0.296	0.296	0.297	0.298	0.299	0.299
1.35	0.300	0.301	0.302	0.302	0.303	0.304	0.305	0.305	0.306	0.307
1.36	0.307	0.308	0.309	0.310	0.310	0.311	0.312	0.313	0.313	0.314
1.37	0.315	0.316	0.316	0.317	0.318	0.318	0.319	0.320	0.321	0.321
1.38	0.322	0.323	0.324	0.324	0.325	0.326	0.326	0.327	0.328	0.329
1.39	0.329	0.330	0.331	0.331	0.332	0.333	0.334	0.334	0.335	0.336
1.40	0.336	0.337	0.338	0.339	0.339	0.340	0.341	0.341	0.342	0.343
1.41	0.344	0.344	0.345	0.346	0.346	0.347	0.348	0.349	0.349	0.350
1.42	0.351	0.351	0.352	0.353	0.353	0.354	0.355	0.356	0.356	0.357
1.43	0.358	0.358	0.359	0.360	0.360	0.361	0.362	0.363	0.363	0.364
1.44	0.365	0.365	0.366	0.367	0.367	0.368	0.369	0.369	0.370	0.371
1.45	0.372	0.372	0.373	0.374	0.374	0.375	0.376	0.376	0.377	0.378
1.46	0.378	0.379	0.380	0.380	0.381	0.382	0.383	0.383	0.384	0.385
1.47	0.385	0.386	0.387	0.387	0.388	0.389	0.389	0.390	0.391	0.391
1.48	0.392	0.393	0.393	0.394	0.395	0.395	0.396	0.397	0.397	0.398
1.49	0.399	0.399	0.400	0.401	0.401	0.402	0.403	0.403	0.404	0.405

TABLE OF NATURAL LOGARITHMS

N	0	1	2	3	4	5	6	7	8	9
1.50	0.405	0.406	0.407	0.407	0.408	0.409	0.409	0.410	0.411	0.411
1.51	0.412	0.413	0.413	0.414	0.415	0.415	0.416	0.417	0.417	0.418
1.52	0.419	0.419	0.420	0.421	0.421	0.422	0.423	0.423	0.424	0.425
1.53	0.425	0.426	0.427	0.427	0.428	0.429	0.429	0.430	0.430	0.431
1.54	0.432	0.432	0.433	0.434	0.434	0.435	0.436	0.436	0.437	0.438
1.55	0.438	0.439	0.440	0.440	0.441	0.441	0.442	0.443	0.443	0.444
1.56	0.445	0.445	0.446	0.447	0.447	0.448	0.449	0.449	0.450	0.450
1.57	0.451	0.452	0.452	0.453	0.454	0.454	0.455	0.456	0.456	0.457
1.58	0.457	0.458	0.459	0.459	0.460	0.461	0.461	0.462	0.462	0.463
1.59	0.464	0.464	0.465	0.466	0.466	0.467	0.468	0.468	0.469	0.469
1.60	0.470	0.471	0.471	0.472	0.473	0.473	0.474	0.474	0.475	0.476
1.61	0.476	0.477	0.477	0.478	0.479	0.479	0.480	0.481	0.481	0.482
1.62	0.482	0.483	0.484	0.484	0.485	0.486	0.486	0.487	0.487	0.488
1.63	0.489	0.489	0.490	0.490	0.491	0.492	0.492	0.493	0.493	0.494
1.64	0.495	0.495	0.496	0.497	0.497	0.498	0.498	0.499	0.500	0.500
1.65	0.501	0.501	0.502	0.503	0.503	0.504	0.504	0.505	0.506	0.506
1.66	0.507	0.507	0.508	0.509	0.509	0.510	0.510	0.511	0.512	0.512
1.67	0.513	0.513	0.514	0.515	0.515	0.516	0.516	0.517	0.518	0.518
1.68	0.519	0.519	0.520	0.521	0.521	0.522	0.522	0.523	0.524	0.524
1.69	0.525	0.525	0.526	0.527	0.527	0.528	0.528	0.529	0.529	0.530
1.70	0.531	0.531	0.532	0.532	0.533	0.534	0.534	0.535	0.535	0.536
1.71	0.536	0.537	0.538	0.538	0.539	0.539	0.540	0.541	0.541	0.542
1.72	0.542	0.543	0.543	0.544	0.545	0.545	0.546	0.546	0.547	0.548
1.73	0.548	0.549	0.549	0.550	0.550	0.551	0.552	0.552	0.553	0.553
1.74	0.554	0.554	0.555	0.556	0.556	0.557	0.557	0.558	0.558	0.559
1.75	0.560	0.560	0.561	0.561	0.562	0.562	0.563	0.564	0.564	0.565
1.76	0.565	0.566	0.566	0.567	0.568	0.568	0.569	0.569	0.570	0.570
1.77	0.571	0.572	0.572	0.573	0.573	0.574	0.574	0.575	0.575	0.576
1.78	0.577	0.577	0.578	0.578	0.579	0.579	0.580	0.581	0.581	0.582
1.79	0.582	0.583	0.583	0.584	0.584	0.585	0.586	0.586	0.587	0.587
1.80	0.588	0.588	0.589	0.589	0.590	0.591	0.591	0.592	0.592	0.593
1.81	0.593	0.594	0.594	0.595	0.596	0.596	0.597	0.597	0.598	0.598
1.82	0.599	0.599	0.600	0.600	0.601	0.602	0.602	0.603	0.603	0.604
1.83	0.604	0.605	0.605	0.606	0.606	0.607	0.608	0.608	0.609	0.609
1.84	0.610	0.610	0.611	0.611	0.612	0.612	0.613	0.614	0.614	0.615
1.85	0.615	0.616	0.616	0.617	0.617	0.618	0.618	0.619	0.620	0.620
1.86	0.621	0.621	0.622	0.622	0.623	0.623	0.624	0.624	0.625	0.625
1.87	0.626	0.626	0.627	0.628	0.628	0.629	0.629	0.630	0.630	0.631
1.88	0.631	0.632	0.632	0.633	0.633	0.634	0.634	0.635	0.636	0.636
1.89	0.637	0.637	0.638	0.638	0.639	0.639	0.640	0.640	0.641	0.641

TABLE OF NATURAL LOGARITHMS

N	0	1	2	3	4	5	6	7	8	9
1.90	0.642	0.642	0.643	0.643	0.644	0.644	0.645	0.646	0.646	0.647
1.91	0.647	0.648	0.648	0.649	0.649	0.650	0.650	0.651	0.651	0.652
1.92	0.652	0.653	0.653	0.654	0.654	0.655	0.655	0.656	0.656	0.657
1.93	0.658	0.658	0.659	0.659	0.660	0.660	0.661	0.661	0.662	0.662
1.94	0.663	0.663	0.664	0.664	0.665	0.665	0.666	0.666	0.667	0.667
1.95	0.668	0.668	0.669	0.669	0.670	0.670	0.671	0.671	0.672	0.672
1.96	0.673	0.673	0.674	0.674	0.675	0.675	0.676	0.677	0.677	0.678
1.97	0.678	0.679	0.679	0.680	0.680	0.681	0.681	0.682	0.682	0.683
1.98	0.683	0.684	0.684	0.685	0.685	0.686	0.686	0.687	0.687	0.688
1.99	0.688	0.689	0.689	0.690	0.690	0.691	0.691	0.692	0.692	0.693
2.00	0.693	0.694	0.694	0.695	0.695	0.696	0.696	0.697	0.697	0.698
2.01	0.698	0.699	0.699	0.700	0.700	0.701	0.701	0.702	0.702	0.703
2.02	0.703	0.704	0.704	0.705	0.705	0.706	0.706	0.707	0.707	0.708
2.03	0.708	0.709	0.709	0.710	0.710	0.710	0.711	0.711	0.712	0.712
2.04	0.713	0.713	0.714	0.714	0.715	0.715	0.716	0.716	0.717	0.717
2.05	0.718	0.718	0.719	0.719	0.720	0.720	0.721	0.721	0.722	0.722
2.06	0.723	0.723	0.724	0.724	0.725	0.725	0.726	0.726	0.727	0.727
2.07	0.728	0.728	0.729	0.729	0.729	0.730	0.730	0.731	0.731	0.732
2.08	0.732	0.733	0.733	0.734	0.734	0.735	0.735	0.736	0.736	0.737
2.09	0.737	0.738	0.738	0.739	0.739	0.740	0.740	0.741	0.741	0.741
2.10	0.742	0.742	0.743	0.743	0.744	0.744	0.745	0.745	0.746	0.746
2.11	0.747	0.747	0.748	0.748	0.749	0.749	0.750	0.750	0.750	0.751
2.12	0.751	0.752	0.752	0.753	0.753	0.754	0.754	0.755	0.755	0.756
2.13	0.756	0.757	0.757	0.758	0.758	0.758	0.759	0.759	0.760	0.760
2.14	0.761	0.761	0.762	0.762	0.763	0.763	0.764	0.764	0.765	0.765
2.15	0.765	0.766	0.766	0.767	0.767	0.768	0.768	0.769	0.769	0.770
2.16	0.770	0.771	0.771	0.771	0.772	0.772	0.773	0.773	0.774	0.774
2.17	0.775	0.775	0.776	0.776	0.777	0.777	0.777	0.778	0.778	0.779
2.18	0.779	0.780	0.780	0.781	0.781	0.782	0.782	0.783	0.783	0.783
2.19	0.784	0.784	0.785	0.785	0.786	0.786	0.787	0.787	0.788	0.788
2.20	0.788	0.789	0.789	0.790	0.790	0.791	0.791	0.792	0.792	0.793
2.21	0.793	0.793	0.794	0.794	0.795	0.795	0.796	0.796	0.797	0.797
2.22	0.798	0.798	0.798	0.799	0.799	0.800	0.800	0.801	0.801	0.802
2.23	0.802	0.802	0.803	0.803	0.804	0.804	0.805	0.805	0.806	0.806
2.24	0.806	0.807	0.807	0.808	0.808	0.809	0.809	0.810	0.810	0.810
2.25	0.811	0.811	0.812	0.812	0.813	0.813	0.814	0.814	0.814	0.815
2.26	0.815	0.816	0.816	0.817	0.817	0.818	0.818	0.818	0.819	0.819
2.27	0.820	0.820	0.821	0.821	0.822	0.822	0.822	0.823	0.823	0.824
2.28	0.824	0.825	0.825	0.825	0.826	0.826	0.827	0.827	0.828	0.828
2.29	0.829	0.829	0.829	0.830	0.830	0.831	0.831	0.832	0.832	0.832

TABLE OF NATURAL LOGARITHMS

N	0	1	2	3	4	5	6	7	8	9
2.30	0.833	0.833	0.834	0.834	0.835	0.835	0.836	0.836	0.836	0.837
2.31	0.837	0.838	0.838	0.839	0.839	0.839	0.840	0.840	0.841	0.841
2.32	0.842	0.842	0.842	0.843	0.843	0.844	0.844	0.845	0.845	0.845
2.33	0.846	0.846	0.847	0.847	0.848	0.848	0.848	0.849	0.849	0.850
2.34	0.850	0.851	0.851	0.851	0.852	0.852	0.853	0.853	0.854	0.854
2.35	0.854	0.855	0.855	0.856	0.856	0.857	0.857	0.857	0.858	0.858
2.36	0.859	0.859	0.860	0.860	0.860	0.861	0.861	0.862	0.862	0.862
2.37	0.863	0.863	0.864	0.864	0.865	0.865	0.865	0.866	0.866	0.867
2.38	0.867	0.868	0.868	0.868	0.869	0.869	0.870	0.870	0.870	0.871
2.39	0.871	0.872	0.872	0.873	0.873	0.873	0.874	0.874	0.875	0.875
2.40	0.875	0.876	0.876	0.877	0.877	0.878	0.878	0.878	0.879	0.879
2.41	0.880	0.880	0.880	0.881	0.881	0.882	0.882	0.883	0.883	0.883
2.42	0.884	0.884	0.885	0.885	0.885	0.886	0.886	0.887	0.887	0.887
2.43	0.888	0.888	0.889	0.889	0.890	0.890	0.890	0.891	0.891	0.892
2.44	0.892	0.892	0.893	0.893	0.894	0.894	0.894	0.895	0.895	0.896
2.45	0.896	0.896	0.897	0.897	0.898	0.898	0.899	0.899	0.899	0.900
2.46	0.900	0.901	0.901	0.901	0.902	0.902	0.903	0.903	0.903	0.904
2.47	0.904	0.905	0.905	0.905	0.906	0.906	0.907	0.907	0.907	0.908
2.48	0.908	0.909	0.909	0.909	0.910	0.910	0.911	0.911	0.911	0.912
2.49	0.912	0.913	0.913	0.913	0.914	0.914	0.915	0.915	0.915	0.916
2.50	0.916	0.917	0.917	0.917	0.918	0.918	0.919	0.919	0.919	0.920
2.51	0.920	0.921	0.921	0.921	0.922	0.922	0.923	0.923	0.923	0.924
2.52	0.924	0.925	0.925	0.925	0.926	0.926	0.927	0.927	0.927	0.928
2.53	0.928	0.929	0.929	0.929	0.930	0.930	0.931	0.931	0.931	0.932
2.54	0.932	0.933	0.933	0.933	0.934	0.934	0.935	0.935	0.935	0.936
2.55	0.936	0.936	0.937	0.937	0.938	0.938	0.938	0.939	0.939	0.940
2.56	0.940	0.940	0.941	0.941	0.942	0.942	0.942	0.943	0.943	0.944
2.57	0.944	0.944	0.945	0.945	0.945	0.946	0.946	0.947	0.947	0.947
2.58	0.948	0.948	0.949	0.949	0.949	0.950	0.950	0.950	0.951	0.951
2.59	0.952	0.952	0.952	0.953	0.953	0.954	0.954	0.954	0.955	0.955
2.60	0.956	0.956	0.956	0.957	0.957	0.957	0.958	0.958	0.959	0.959
2.61	0.959	0.960	0.960	0.960	0.961	0.961	0.962	0.962	0.962	0.963
2.62	0.963	0.964	0.964	0.964	0.965	0.965	0.965	0.966	0.966	0.967
2.63	0.967	0.967	0.968	0.968	0.969	0.969	0.969	0.970	0.970	0.970
2.64	0.971	0.971	0.972	0.972	0.972	0.973	0.973	0.973	0.974	0.974
2.65	0.975	0.975	0.975	0.976	0.976	0.976	0.977	0.977	0.978	0.978
2.66	0.978	0.979	0.979	0.979	0.980	0.980	0.981	0.981	0.981	0.982
2.67	0.982	0.982	0.983	0.983	0.984	0.984	0.984	0.985	0.985	0.985
2.68	0.986	0.986	0.987	0.987	0.987	0.988	0.988	0.988	0.989	0.989
2.69	0.990	0.990	0.990	0.991	0.991	0.991	0.992	0.992	0.993	0.993

TABLE OF NATURAL LOGARITHMS

N	0	1	2	3	4	5	6	7	8	9
2.70	0.993	0.994	0.994	0.994	0.995	0.995	0.995	0.996	0.996	0.997
2.71	0.997	0.997	0.998	0.998	0.998	0.999	0.999	1.000	1.000	1.000
2.72	1.001	1.001	1.001	1.002	1.002	1.002	1.003	1.003	1.004	1.004
2.73	1.004	1.005	1.005	1.005	1.006	1.006	1.006	1.007	1.007	1.008
2.74	1.008	1.008	1.009	1.009	1.009	1.010	1.010	1.011	1.011	1.011
2.75	1.012	1.012	1.012	1.013	1.013	1.013	1.014	1.014	1.015	1.015
2.76	1.015	1.016	1.016	1.016	1.017	1.017	1.017	1.018	1.018	1.018
2.77	1.019	1.019	1.020	1.020	1.020	1.021	1.021	1.021	1.022	1.022
2.78	1.022	1.023	1.023	1.024	1.024	1.024	1.025	1.025	1.025	1.026
2.79	1.026	1.026	1.027	1.027	1.027	1.028	1.028	1.029	1.029	1.029
2.80	1.030	1.030	1.030	1.031	1.031	1.031	1.032	1.032	1.032	1.033
2.81	1.033	1.034	1.034	1.034	1.035	1.035	1.035	1.036	1.036	1.036
2.82	1.037	1.037	1.037	1.038	1.038	1.039	1.039	1.039	1.040	1.040
2.83	1.040	1.041	1.041	1.041	1.042	1.042	1.042	1.043	1.043	1.043
2.84	1.044	1.044	1.045	1.045	1.045	1.046	1.046	1.046	1.047	1.047
2.85	1.047	1.048	1.048	1.048	1.049	1.049	1.049	1.050	1.050	1.050
2.86	1.051	1.051	1.052	1.052	1.052	1.053	1.053	1.053	1.054	1.054
2.87	1.054	1.055	1.055	1.055	1.056	1.056	1.056	1.057	1.057	1.057
2.88	1.058	1.058	1.058	1.059	1.059	1.060	1.060	1.060	1.061	1.061
2.89	1.061	1.062	1.062	1.062	1.063	1.063	1.063	1.064	1.064	1.064
2.90	1.065	1.065	1.065	1.066	1.066	1.066	1.067	1.067	1.067	1.068
2.91	1.068	1.068	1.069	1.069	1.070	1.070	1.070	1.071	1.071	1.071
2.92	1.072	1.072	1.072	1.073	1.073	1.073	1.074	1.074	1.074	1.075
2.93	1.075	1.075	1.076	1.076	1.076	1.077	1.077	1.077	1.078	1.078
2.94	1.078	1.079	1.079	1.079	1.080	1.080	1.080	1.081	1.081	1.081
2.95	1.082	1.082	1.082	1.083	1.083	1.083	1.084	1.084	1.085	1.085
2.96	1.085	1.086	1.086	1.086	1.087	1.087	1.087	1.088	1.088	1.088
2.97	1.089	1.089	1.089	1.090	1.090	1.090	1.091	1.091	1.091	1.092
2.98	1.092	1.092	1.093	1.093	1.093	1.094	1.094	1.094	1.095	1.095
2.99	1.095	1.096	1.096	1.096	1.097	1.097	1.097	1.098	1.098	1.098
3.00	1.099	1.099	1.099	1.100	1.100	1.100	1.101	1.101	1.101	1.102
3.01	1.102	1.102	1.103	1.103	1.103	1.104	1.104	1.104	1.105	1.105
3.02	1.105	1.106	1.106	1.106	1.107	1.107	1.107	1.108	1.108	1.108
3.03	1.109	1.109	1.109	1.110	1.110	1.110	1.111	1.111	1.111	1.112
3.04	1.112	1.112	1.113	1.113	1.113	1.114	1.114	1.114	1.114	1.115
3.05	1.115	1.115	1.116	1.116	1.116	1.117	1.117	1.117	1.118	1.118
3.06	1.118	1.119	1.119	1.119	1.120	1.120	1.120	1.121	1.121	1.121
3.07	1.122	1.122	1.122	1.123	1.123	1.123	1.124	1.124	1.124	1.125
3.08	1.125	1.125	1.126	1.126	1.126	1.127	1.127	1.127	1.128	1.128
3.09	1.128	1.128	1.129	1.129	1.129	1.130	1.130	1.130	1.131	1.131

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 7

COMPONENTS OF LIQUID SYSTEMS

HEAT STORAGE

CONTROLS

HEAT EXCHANGERS

PUMPS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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OBJECTIVE

The objective in this module is to present information on components of liquid systems so that participants will be able to:

1. Select the appropriate size storage unit.
2. Properly size pumps.
3. Select heat exchangers.
4. Install a control unit for the system.

INTRODUCTION

The principal components of liquid-type solar systems are the collectors, heat storage units, pumps, and heat exchangers which are interconnected with pipes and regulated by automatic controls. Collectors were described in a previous module and specific attention is devoted here to other components of the solar system.

HEAT STORAGE

Storage of solar thermal energy is required in solar heating and cooling systems so that solar heat can be delivered to the building during non-sunny periods. The volume of storage needed is principally dependent on collector area. In liquid-based systems where solar heat is generally first collected in storage before delivery to the load, the storage unit should be large enough for all-day solar heat collection without penalizing collector efficiency, and small enough to raise the temperature in storage to meet load requirements.

Heat may be stored by raising the temperature of the storage medium (sensible heat storage), changing the state of the medium from solid to liquid (latent heat storage), or by chemical reaction of a suitable medium. Heat may be recovered by lowering the temperature of the storage medium, changing the phase from liquid to solid or by reversing the chemical reaction. For solar heating and cooling systems, sensible heat storage in water is commonly used. Some phase-change materials (PCM) are commercially available, but as yet there has been little practical experience using PCM storage units in solar heating and cooling systems. Chemical heat storage is appropriate for high temperature systems and is not being actively considered for space heating and cooling systems.

WATER STORAGE

The specific heat of water is 1 Btu/(lb·°F) which means that one Btu of heat energy can be stored in one pound of water when the temperature of the water is raised one degree Fahrenheit. One gallon (U.S.) of water can store about 8.25 Btu for each degree of temperature rise (between 100°F to 160°F) or 495 Btu can be stored when the temperature is raised from 100°F to 160°F. A water tank with 1000 gallons of water will thus store 495,000 Btu of heat energy when the temperature is raised from 100°F to 160°F, which is a typical operating temperature range for water storage tanks in liquid-type solar heating systems.

Recommended Water Volume

Precise determination of water storage volume is not required for effective system operation, but there is a practical range of 1.5 to

2.5 gallons of water storage for each square foot of collector area. Thus, for a system with a collector area of 400 ft², storage volume should be in the range of 600 to 1000 gallons. The effect of storage volume on the annual fraction of the total heating load supplied by the solar system is illustrated in Figure 7-1.

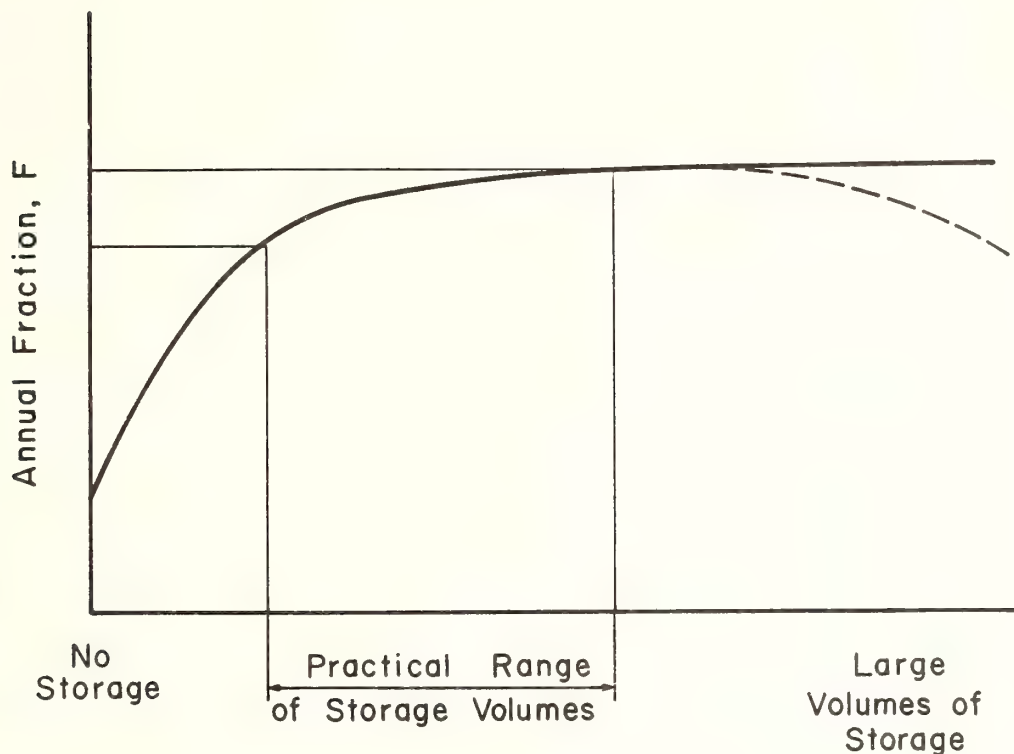


Figure 7-1. Effect of Storage Size on Solar Heat Contribution to Total Load

With no storage the amount of solar heat provided is limited because solar heat could be delivered and used only during the day when the thermostat calls for heat. The contribution of solar heat to the load increases rapidly with storage size, and the lower limit of the practical range is the knee in the curve. Theoretically, as the storage volume continues to increase, the solar fraction should also increase.

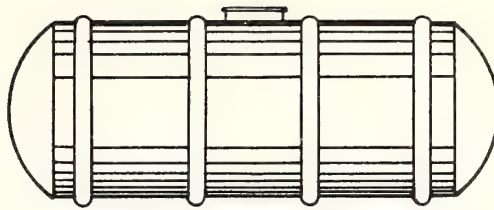
However, the maximum temperature achieved in storage reduces with increases in size, and with lower temperatures less useful heat is delivered to the load. Storage volume should not be oversized for solar heating and cooling systems.

Water Tanks

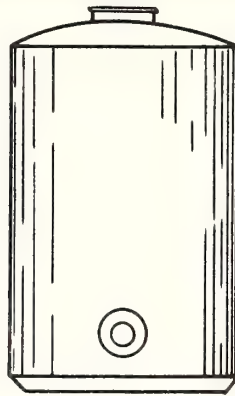
There are several types of water tanks that can be used in solar systems. Steel and fiber-reinforced plastic (FRP) tanks of the type shown in Figure 7-2 are commonly used, with steel tanks being more appropriate for higher water temperatures. With FRP tanks, even those that are especially designed for high temperature liquids, there should be controls to limit the water temperature in the tank.

Tanks made of concrete or masonry blocks and lined with waterproof liners may also be used, as illustrated in Figure 7-3. It may be convenient to utilize part of the building foundation walls to form one or more sides of the tank. The foundation for concrete tanks is of special concern because settlement can cause cracks to develop and the waterproof liner can be damaged if the cracks are large. Hypalon or butyl rubber liners are especially suitable for such installations.

A prefabricated modular tank of rectangular shape, as shown in Figure 7-4, is commercially available and is suitable for storage of water. The tank is made of flat sections with foam insulation sandwiched between two thin galvanized steel plates. Special connectors, corner sections, and steel channel whalers are used to form a structurally sound box, the time required for assembly of which is only a few hours. The preinsulated tank is especially well suited for retrofit systems because pieces can be easily carried through doorways. A waterproof liner is required.



UNDERGROUND STORAGE TANK



ABOVE-GROUND STORAGE TANK

Figure 7-2. Cylindrical Tanks

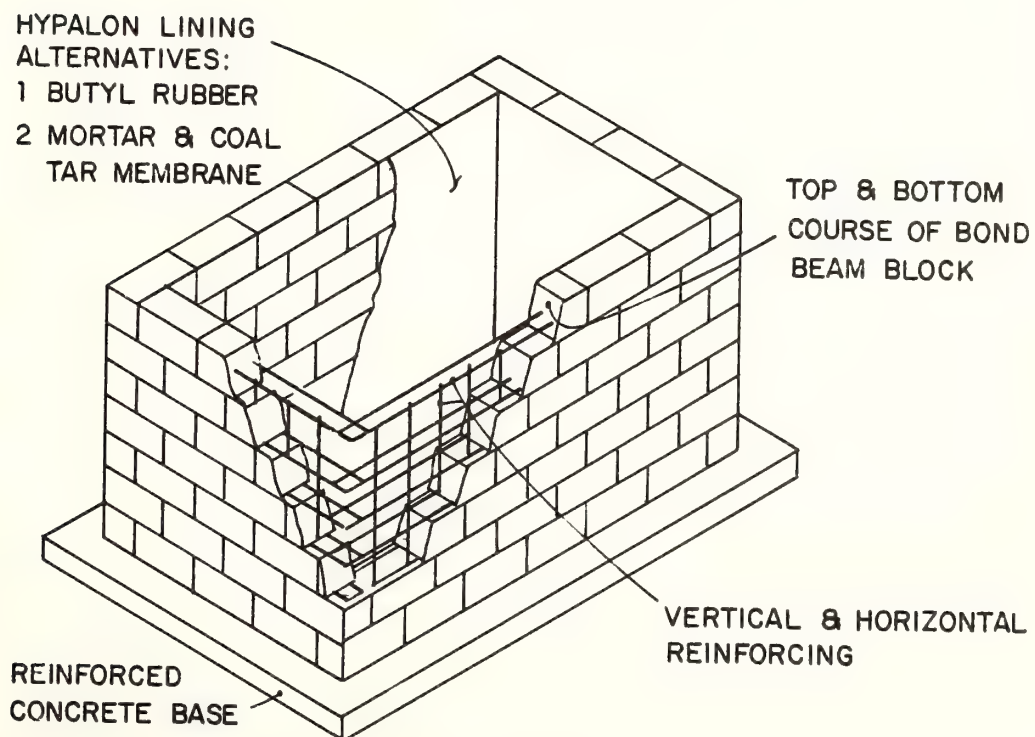


Figure 7-3. Reinforced Concrete Block Tank

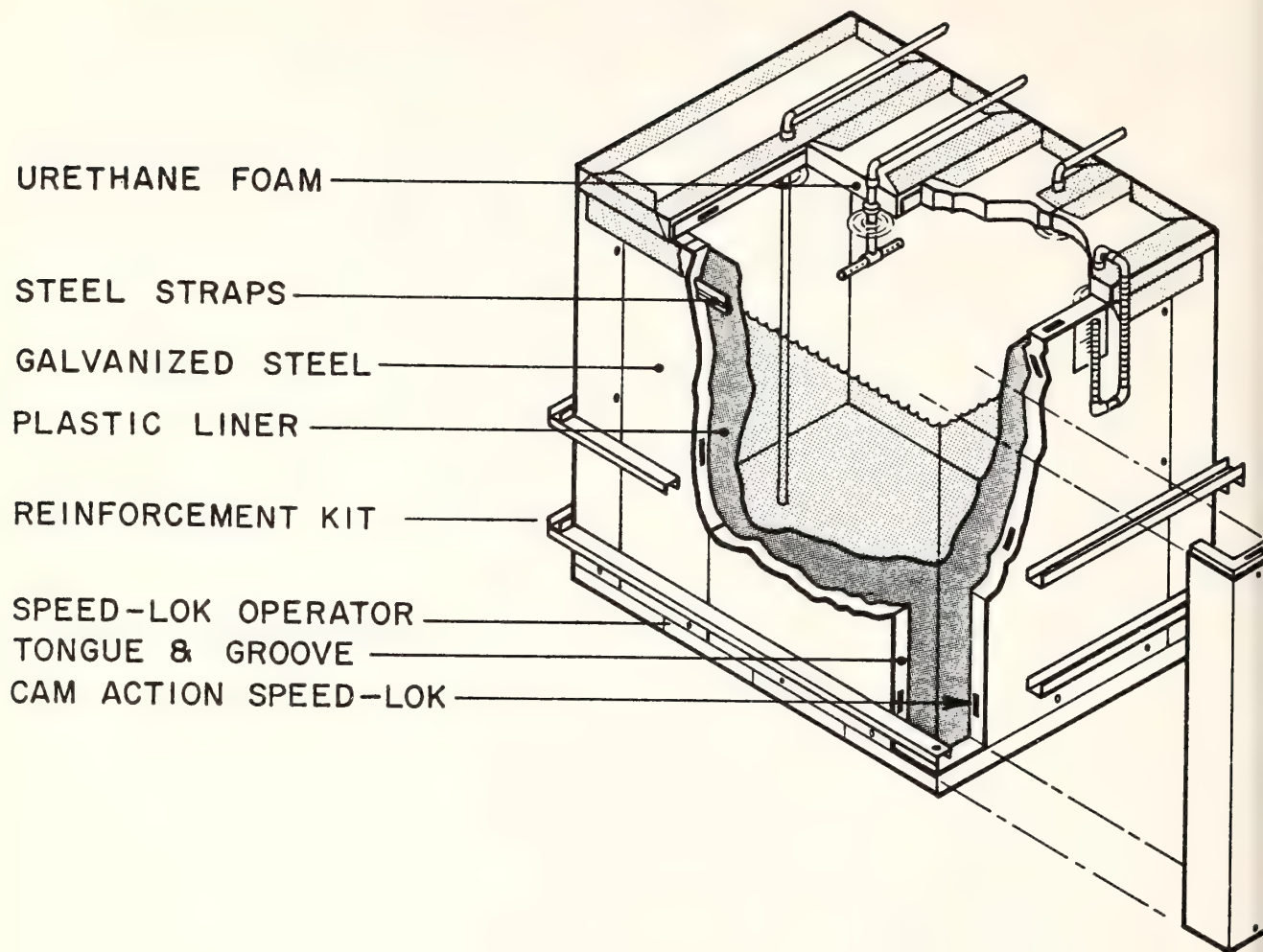


Figure 7-4. A Prefabricated Water Tank For Thermal Energy Storage

Storage Location

Water tanks may be located below or above ground level, and may be inside or outside the building. It is strongly recommended that heat storage tanks be placed within the building enclosure for ease of installation and maintenance. An indoor storage location also protects the vessel and insulation from weathering. When storage tanks are located indoors, heat losses from the tank can meet part of the heat requirements of the building. To avoid overheating in summer, heat

storage should normally not be in use. Another arrangement is to locate the storage tank in a room where heat can be vented outdoors during the summer and communicated to the interior space during the winter.

Installation

Storage tanks are normally the first component of a solar system to be installed. Because of their size, it is advantageous to install the tank before the building is enclosed (for new construction), but the tank should also be placed where removal and replacement are possible. Sectional tanks discussed in the previous section are particularly suitable for installation in existing buildings.

Storage tanks should be well insulated on all surfaces with insulation that rates to about R-30. For cylindrical tanks the bottom and saddle, or leg supports, are the most difficult to insulate. If a flat-bottom tank is installed, rigid insulation may be used below it as shown in Figure 7-5. However, the insulation should be supported above the floor so that it will not become wet when there is standing water on the floor. The prefabricated and preinsulated tank shown in Figure 7-4 can be supported on a base placed on the floor so that the bottom of the tank will be dry.

When storage tanks are placed underground, the insulation around the tank must be of a type which will not absorb moisture. Materials such as neoprene foam could be used. The tank should be placed below the frost line unless a concrete-lined pit is provided. Underground installations have not been very successful in practice and should

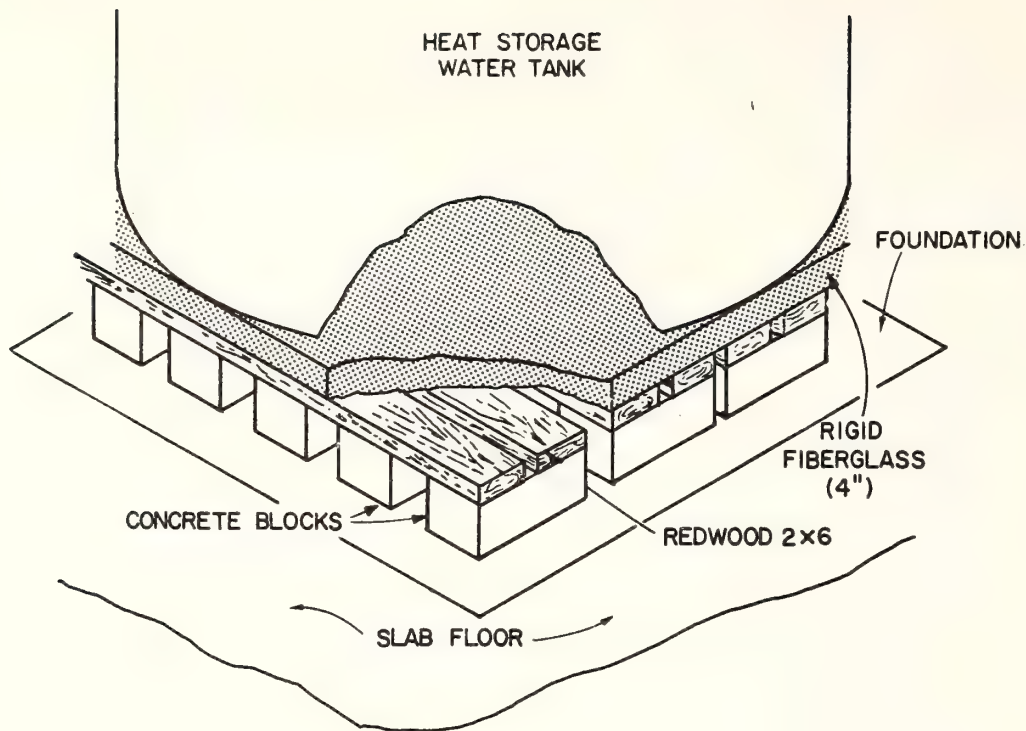


Figure 7-5. Bottom Insulation and Support Scheme for Flat-Bottom Water Storage Tanks

be avoided if possible. Whether storage tanks are placed underground or in the building, the tank and piping should be leak-tested after assembly.

Special Provisions

There should be provisions for draining water tanks. Steel tanks are generally provided with fittings for gravity drainage, but commercial FRP tanks may not have suitable openings. Special connections are available for installation on FRP tanks. Piping connections for tanks with liners are usually made through the lid because penetrations through the liner are sources for leakage. To drain such tanks, the water must be pumped out.

Connections of dissimilar metals to steel tanks should be avoided to prevent galvanic corrosion. When copper pipes are to be attached to steel tanks, it is advisable to use neoprene or silicone rubber hose between steel pipe stubs and copper piping. Special nylon or teflon-lined pipe connections are available.

Water will expand and contract with temperature changes, so an open vent should be provided to prevent storage tanks from becoming pressurized. For most residential-sized tanks, a 2-in diameter pipe vent should be adequate. There will be vapor loss through the vent and there should be provisions for adding make-up water. Water can be added from a supply pipe with manual valve control, using a sight-glass to observe the water level in the tank, or a float control valve may be installed in the supply line to maintain a minimum water level.

Boiling can occur in the storage tank and a vent will prevent damage to the system, the tank, and the contents of the building. While it is an uncommon occurrence during the heating season, boiling may be expected during the summer while the system is used only for domestic water heating. Frequent boiling will also cause build-up of mineral deposits with consequent corrosion, so water softeners are recommended for removal of minerals from the storage water. Make-up water should also be treated.

In some systems, unvented pressurized tanks are used, particularly in conjunction with hydronic distribution piping and hot water boilers for auxiliary heat supply. Pressurized expansion tanks then must also be used to accommodate the changing liquid volumes. If boiling occurs, a pressure relief valve permits steam to escape through a vent.

A high temperature shut-off control switch for the collector pump may be used with some collectors to avoid boiling in storage. In such designs, boiling must be permitted in the collectors, or they must be drained, and the collectors must be able to sustain the high stagnation temperatures that will be reached with no fluid flowing through the collectors.

Corrosion is a potential problem whenever water is contained in a steel tank, and the probability of corrosion greatly increases as the temperature rises. Various types of lining will increase the life of the steel tank but will add substantially to the cost. Removal of minerals with a water softener and corrosion inhibitors in storage water are recommended whenever steel tanks are used.

Water leakage is to be avoided, but some leakage from pipe fittings is inevitable. Floor drains should therefore be provided near the tank.

PHASE-CHANGE STORAGE

Large amounts of solar heat can be stored in special materials such as salt hydrates with a change in phase from a solid to a liquid state. When the solid material is heated and melting temperature is reached, a large quantity of heat is absorbed during the melting process. Further heating raises the temperature of the liquid phase and more heat is stored. The process is illustrated in Figure 7-6.

During the reverse cycle, when the temperature of the storage medium drops, heat is recovered from the liquid, and when the freezing point is reached, a large quantity of heat is released as the liquid changes to a solid. A very small temperature difference is sufficient to change the phase of the material from solid to liquid or liquid to solid state.

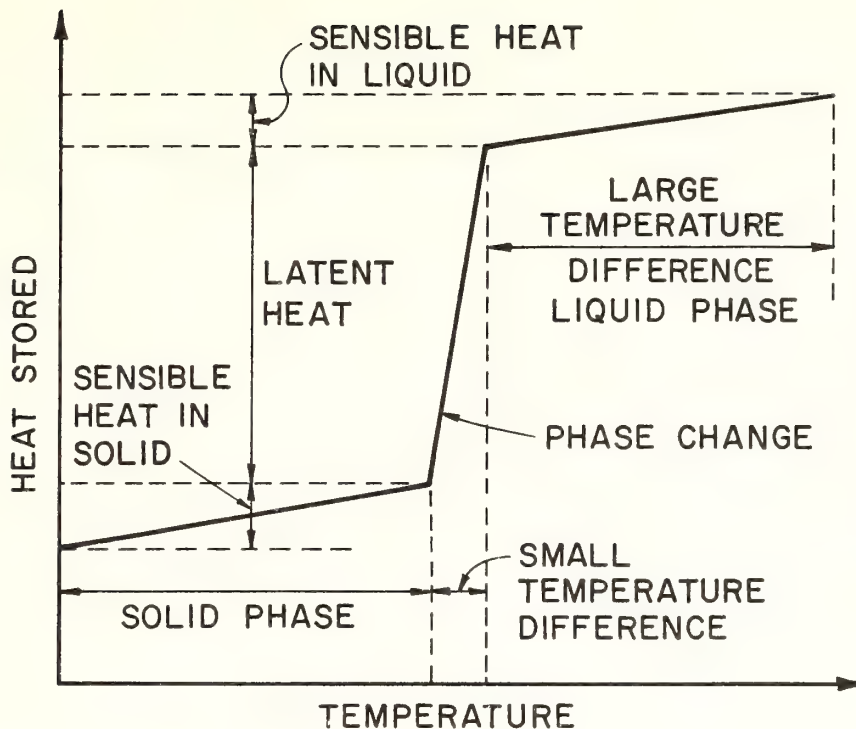


Figure 7-6. Heat Storage in Phase-Change Materials

There are two principal advantages to use of latent heat storage materials. The first is that it takes less volume to store a given quantity of heat compared to water storage, and the second is that the temperature remains nearly constant during phase change. A constant storage temperature, if relatively low, is conducive to good collector efficiency. There are, however, some disadvantages with latent heat storage units. The useful service life is generally limited, although significant improvements are being made. The melting temperatures of suitable materials, like Glaubers salt, are too low for effective use in space heating. (Glaubers salt melts at about 90°F.) Currently PCM materials are expensive when compared to water storage units, and containerizing is costly also. Lastly, a current handicap is limited practical experience with PCM materials in solar heating systems.

SYSTEM CONTROLS

Controls in solar systems must: (1) regulate the automatic collection and distribution of solar heat, and (2) operate the conventional heating system in conjunction with the solar system. Properly installed controls will maximize solar energy collection and distribution and minimize electrical energy consumption to operate the system. Designing controls is a specialized field and is beyond the scope of this manual and training course, but it is important that basic functions of controls and principles of proper installation be understood.

BASIC COMPONENTS

A block diagram of the three basic components of a control system is shown in Figure 7-7. The sensors are generally temperature-measuring



Figure 7-7. Basic Components of Controls

devices although optical sensors may be involved in certain systems. Comparators are differential thermostats which determine the difference in temperatures of two sensors. When the difference in temperatures is greater or less than preset values, output devices respond according to a prearranged plan. The output devices are the mechanical components of

the system, which in liquid systems are pumps, motorized valves, circulating fans or fan-coil units and auxiliary heaters.

Although there are several types of controllers, the most common type is an "on-or-off" system where the mechanical device, such as a pump, is activated or stopped depending upon open or closed electrical circuits. Another type of control system regulates the speed of the pump (or selects an appropriate stage of a multiple-speed pump motor) according to the temperature difference of two sensors. This type of controller is yet uncommon, and experiments are being conducted to determine if the benefits of sophistication in controls are sufficient to overcome the disadvantages of greater cost and complexity.

PRINCIPLES OF OPERATION

Collecting Solar Heat

A schematic diagram of a liquid-type solar heating system is illustrated in Figure 7-8. The system consists of flat-plate liquid-heating collectors with an antifreeze solution in the "collector loop" which is circulated by pump No. 1. A shell-and-tube type liquid-to-liquid heat exchanger is used to transfer heat to water in the storage tank, circulated by pump No. 2. Hot water from storage is circulated to a load heat exchanger by pump No. 3 to heat the building space. Two solenoid-operated valves are required; valve No. 1, which is normally open, controls water flow from the solar storage tank, and valve No. 2, which is normally closed, controls circulation through the auxiliary heater loop.

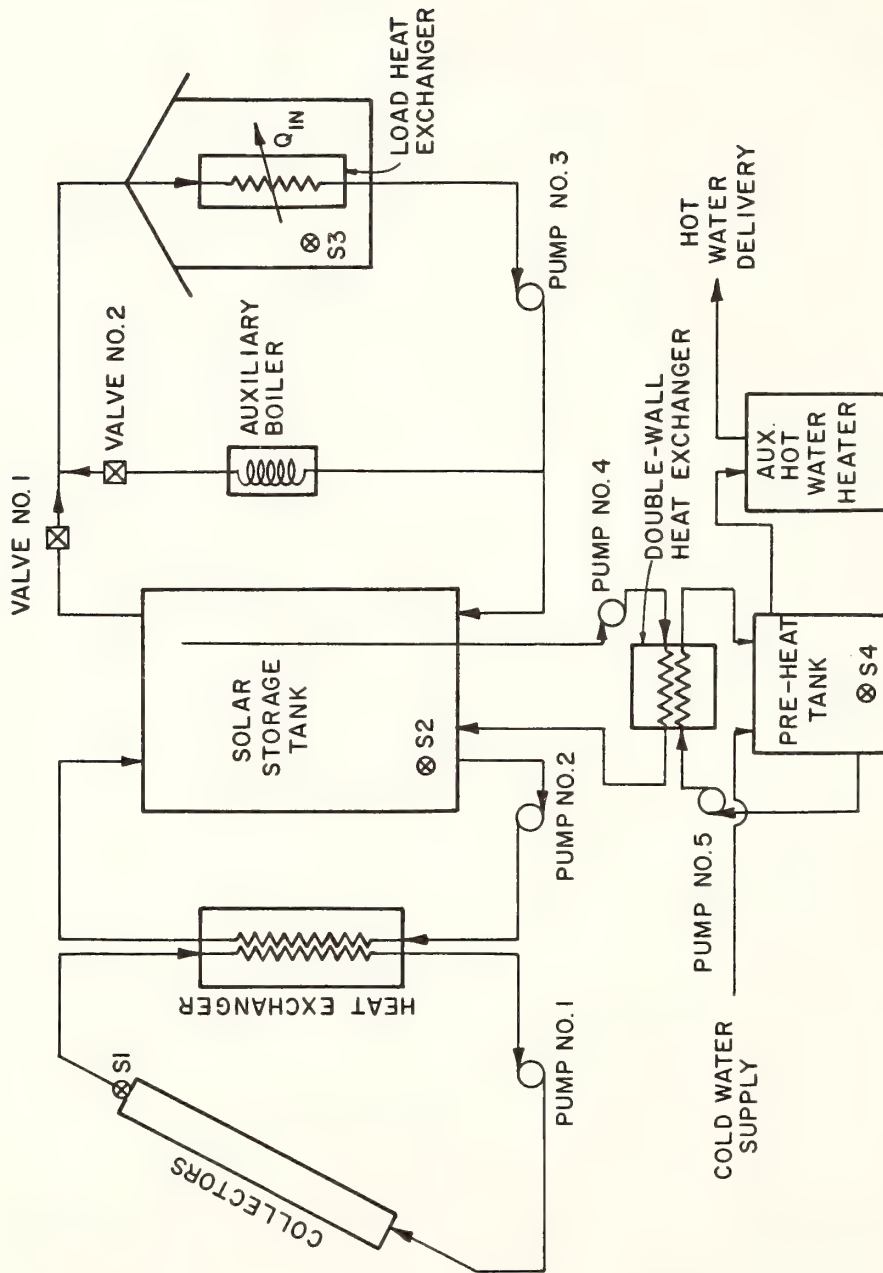


Figure 7-8. Sensor Locations in a Typical Liquid-Heating System

Domestic hot water is preheated with solar heated water from the storage tank. To separate the non-potable water in storage from the potable water in the preheat tank, a double-wall heat exchanger is used. Pumps No. 4 and No. 5 circulate the storage and domestic water through separate pipe loops.

There are three temperature sensors identified in Figure 7-8. Sensor S1 is located near the top of one collector in the array, S2 is located near the bottom of the storage tank and S4 is located near the bottom of the pre-heat tank. Sensor S3 is a two-stage thermostat located in the heated space.

Typical variations in temperature of the liquids at collector exit and in storage are shown on Figure 7-9. The solid curve shows the storage temperature as sensed by S2 and the dashed line is the collector temperature at S1. The collector fluid temperature is low in the

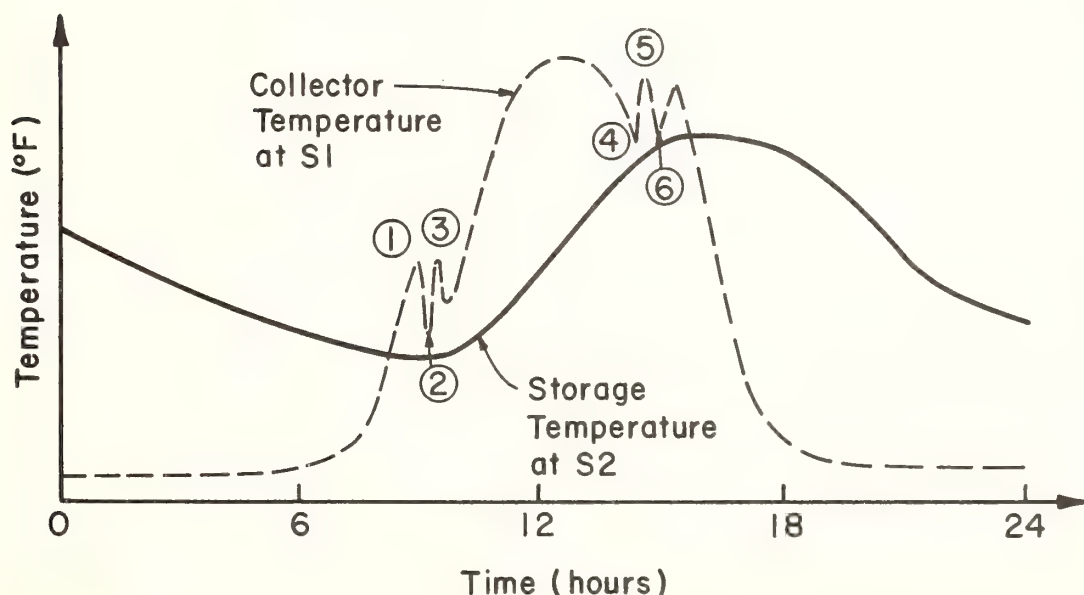


Figure 7-9. Typical Temperature Profiles of Collector and Storage Liquids in a Liquid-Heating System

morning but starts to rise as soon as solar radiation is absorbed in the collector. At about 0830 hrs the collector temperature exceeds the storage temperature, and because there is no flow through the collector, the temperature rises rapidly. When the collector temperature exceeds the storage temperature by a preset amount, typically 20°F (at point ①), pumps 1 and 2 are activated simultaneously by the controller. When liquid circulation begins, a surge of cold liquid in the pipes moves through the collectors. Because the intensity of solar radiation on the collectors is low in the morning, the cold liquid does not heat up rapidly and the temperature at S1 drops to point ②. If the temperature difference between S1 and S2 drops to another preset value, typically 3°F, pumps 1 and 2 deactivate. With no circulation, the liquid in the collector is heated rapidly, the temperature at S1 rises to point ③, and pumps 1 and 2 are again activated. As the liquid in the pipe loop is still cool, the temperature at S1 drops again as circulation commences. The number of on-off cycles at system start-up in the morning depends on solar intensity, setting of the differential thermostats, fluid flow rate, and volume of water in the collector loop. Generally, on-off cycling should be minimized to reduce wear of pump motor and starter relay contacts. After two or three cycles at the beginning, the fluid temperature at S1 will continue to increase as illustrated in the figure and the pumps will remain on.

After mid-day the temperature at S1 decreases but storage temperature continues to rise until mid-afternoon. When the temperature difference between S1 and S2 decreases to shut-off point ④, pumps 1 and 2 deactivate. After circulation stops, S1 will rise again because solar energy is still being received. The difference in temperature

between S1 and S2 again reaches 20°F and the pumps are activated at point ⑤ . However, the intensity of solar energy is not sufficient to maintain high temperature at S1 and the pumps shut off at point ⑥ . The number of cycles which may occur at the end of the day depends on control settings and fluid flow rates, just as it does at the beginning of the day.

Control of Space Heating

Fluid circulation through the load heat exchanger is controlled by thermostat, S3. When the room cools and the first stage contact is made, pump 3 is activated. Valve 1 is normally open and valve 2 is normally closed, so that water from the storage tank is circulated and solar-heated water is first used to heat the rooms. If the solar-heated water in storage is not sufficiently warm to meet the heating demand, the room air continues to cool down and the second stage is contacted. Valve 1 closes, valve 2 opens and simultaneously the auxiliary boiler is activated. Since the auxiliary boiler size is adequate to supply the heating load for the coldest night during the winter, the room air will be heated. When the room air temperature rises beyond the thermostat set point by about 1°F, the auxiliary heater and pump 3 shut off.

In the control scheme described above, the room air temperature will drop 2°F to 3°F to activate the first stage and another 2°F to 3°F to reach second stage. Hysteresis adds another 1°F to 2°F above the set point so that when solar heat in storage has been depleted, a temperature "swing" as large as 8°F can occur (64°F to 72°F) in the rooms.

To minimize the temperature swing the thermostat can be adjusted so that the temperature differences between stages and set point are reduced to 1°F to 2°F. However, when the temperature differences are too small, frequent cycling may occur in the load loop. An alternative strategy to reduce temperature swing in the rooms is to limit use of storage water above a set temperature, say 100°F. When the first stage is contacted and the storage temperature is above 100°F, solar-heated water is circulated to the load heat exchanger, but if storage water temperature is below 100°F, the auxiliary heater is activated at the first stage contact, preventing further cooling of the rooms.

The advantage of the latter control strategy is reduction in temperature variations in the rooms. A disadvantage is limitation in use of the solar heat in storage since there may be many occasions when water temperature less than 100°F is adequate to supply heat to the rooms.

Control of Domestic Water Heating

A differential thermostat is normally used to control the pre-heating of domestic water. When the temperature difference between S2 and S4 is greater than a set point, say 20°F, pumps 4 and 5 are activated and water is circulated through the respective loops. Heat is exchanged from the solar-heated water in the storage tank to water in the pre-heat tank. As water in the pre-heat tank is warmed, the temperature difference decreases and the pumps shut off when a lower set point is reached, say at 3°F.

With selection of proper pump sizes and heat exchangers, a significant portion of the DHW load can be supplied by solar energy

because solar heat in the main storage tank can be used to pre-heat domestic water even when the main tank temperature is less than 100°F. During the evening hours and at night when use of hot water increases, the main thermal storage tank will be significantly warmer than the cold water temperature entering the pre-heat tank from the city mains.

COMPONENTS

Differential Thermostat

Sensors used in liquid systems are typically thermistors, and matching sensors are usually provided with the controller. A typical circuitry for a single-function differential thermostat (controller) is shown in Figure 7-10. Temperature sensors are connected to a

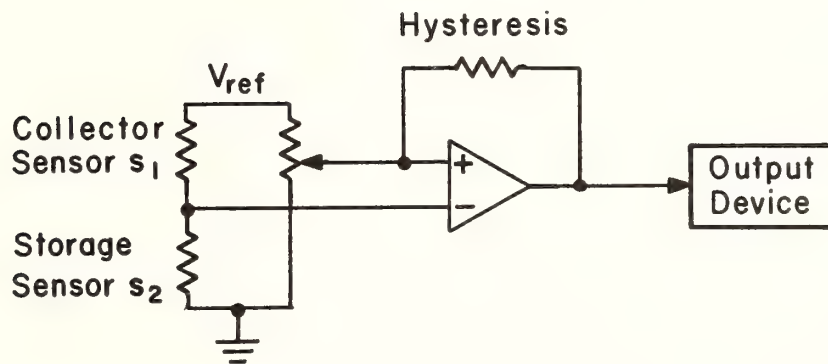


Figure 7-10. Typical Circuitry for a Differential Thermostat

comparator and to an output device. Hysteresis is the range in temperature between the start and stop set points. A pictorial representation of hysteresis is shown in Figure 7-11, which is achieved electrically by the feedback loop shown in Figure 7-10. As the temperature difference increases and eventually reaches an upper set point, ΔT_{ON} , the electrical circuit to the output device is completed and the device is turned

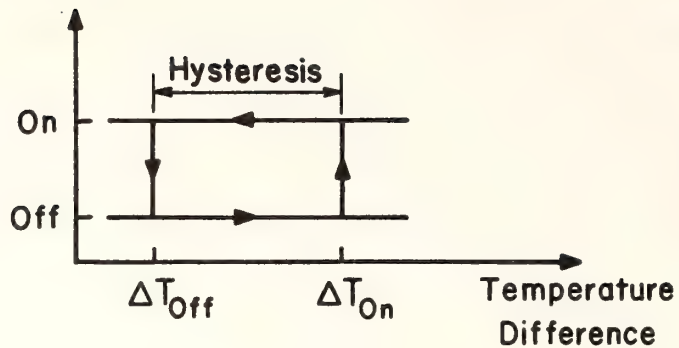


Figure 7-11. Pictorial Representation of Hysteresis in the Differential Thermostat

on. When the temperature difference decreases to the point where it is equal to ΔT_{OFF} , the electrical circuit is broken and shuts off the output device. To minimize on-off cycling, the ratio of on to off temperature differences should be 5 to 7. In the preceding example, the starting temperature difference was 20°F and the stopping temperature difference was 3°F. The ratio is slightly less than seven and is satisfactory. A larger ratio will delay starting time and a smaller ratio will cause cycling.

Room Thermostat

A thermostat with two-stage heating control is recommended for residential solar heating systems (with one-stage control for the cooling system if necessary). Various thermostat designs feature, "on", "off", or "automatic" fan control to circulate the room air, with "heat", "cool", or "automatic" switches for heating and cooling functions. Usually the thermostat is the only control with which the occupant needs to be concerned. Once set to winter or summer comfort levels

no further manual intervention is necessary unless the occupant wishes to change room temperature.

Thermostats should be installed at locations where average room temperatures are sensed. Instructions are normally supplied by the manufacturer.

Temperature Sensors

There are many types of temperature sensors that can be used in the control subsystem, such as thermocouples, thermistors, silicon transistors, bimetallic elements, and liquid or vapor expansion units. Liquid or vapor expansion units are seldom used because other temperature sensors are cheaper and dependable. Thermocouples are frequently used for temperature measurement but are not often used in controls because the voltage output is low, in the millivolt range, and without amplification the voltage is insufficient for reliable use in controls. Thermistors and silicon transistors are commonly used because the voltage outputs from these sensors are high (3-10 volts).

INSTALLATION OF CONTROL HARDWARE

Control Panels

Except for temperature sensors, components of controls are generally packaged in a compact control box or panel. Prewired units are provided with lugs for attaching wires leading to temperature sensors, motors, and valves. Power for the control panel will usually be household, 115-volt, single-phase AC power, which is stepped down to 24 volts for room thermostats, and control relays which activate motors and valves. An instruction manual should be provided with the controller.

Locations of Temperature Sensors

Locations of temperature sensors are important, and there are some preferred locations. The sensor which measures fluid temperature at the collector exit should be located in the outlet pipe from a collector in an upper row of an array. It is preferred that the sensor be in a wet well, but it is acceptable for the sensor to be in a dry well provided there is good thermal contact of the sensor with the wall of the dry well. Both wet and dry wells should be well insulated. The collector sensor should be placed as near to the outlet as possible so that it can register the temperature of the collector fluid when it is not circulating.

The sensor in the storage tank should be located near the bottom because the coldest water will be at the lower levels. The sensor in the hot water pre-heat tank should also be located near the bottom. If a temperature limiter is used, the sensor should be located near the top of the tank.

AUXILIARY HEAT CONTROL

Conventional boilers and furnaces are activated in conjunction with the pumps and blowers in the solar system. The second stage thermostat is generally the main control for activating an auxiliary heater.

CONTROL SYSTEM CHECK-OUT

The control system should be checked after installation with a "dry" run through the full sequence of modes. The room thermostat can usually be manipulated to require heating or cooling, and temperature sensor terminals can be "shorted" to represent high temperature to start

a heat collection mode or domestic water pre-heating mode. A pre-operational check-out will assure that the system will "work" when it is put into operation. Adjustments to the control system may sometimes be necessary to achieve high performance of the system. A common adjustment is resetting of "on" and "off" set points to start and stop solar heat collection at proper times of the day.

HEAT EXCHANGERS

COLLECTOR TO STORAGE HEAT EXCHANGER

Shell-and-Tube Type

A heat exchanger is used to transfer heat from collector fluid to storage fluid. Among several designs for heat exchangers a shell-and-tube type, shown schematically in Figure 7-12, is the simplest.

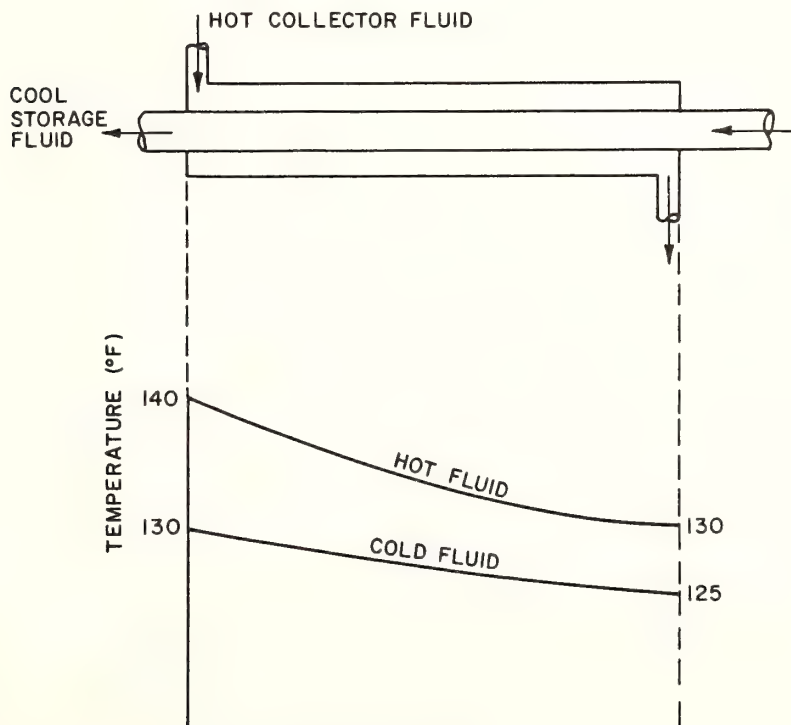


Figure 7-12. Single-Pass Counterflow Shell-and-Tube Heat Exchanger

A tube which passes either the cold or hot fluid, shown as cold fluid in Figure 7-12, is placed concentrically inside a larger tube which forms the shell. The hot and cold fluids may flow in the same direction (parallel flow) or in opposite directions (counterflow) as shown. It has been assumed in this case that the cold fluid is being circulated at twice the flow rate of the hot fluid, so its temperature changes only half as many degrees. A counterflow type is appropriate for solar systems to minimize the temperature difference between storage and collector exit fluid temperatures. In parallel flow heat exchangers, the cold fluid can never be warmer than the hot fluid, while with counterflow types the exit temperature of the cold fluid can be higher than the exit temperature of the hot fluid (shown equal in the figure). A counterflow design also requires less surface area to transfer heat at a given rate than a parallel type.

A more practical design for a shell-and-tube heat exchanger involves multiple tubes within a shell as shown in Figure 7-13. There are baffles within the shell which force the shell side flow to move in a serpentine path across the baffles as shown by the arrows. With multiple tubes, there is a larger surface area for heat exchange within a given shell diameter, enabling a shorter length of heat exchanger to

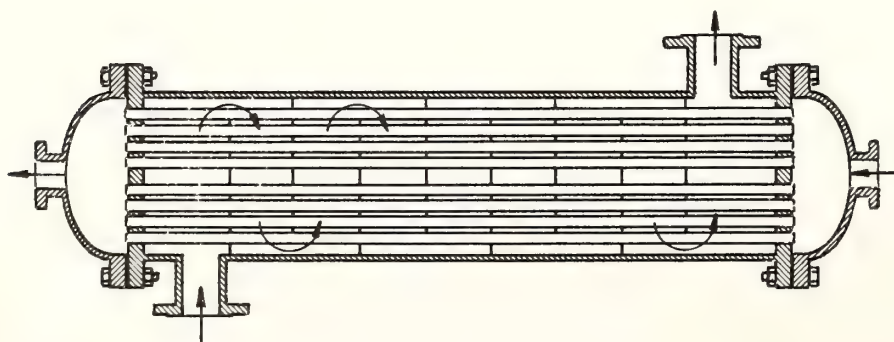


Figure 7-13. Multiple-Tube Heat Exchanger

be used as compared to single-tube design. There are also multiple-pass heat exchanger designs which cause flow in the tubes to reverse directions within the shell.

Selection

The rate of heat transfer from the hot fluid to the cold fluid, Q_{HX} , in a shell-and-tube heat exchanger may be written as

$$Q_{HX} = (\dot{m}c_p)_c(T_1 - T_2), \text{ (Btu/hr)} \quad (7-1)$$

where

$(\dot{m}c_p)_c$ is the heat-capacity flow rate of the collector liquid, Btu/(hr·°F),

\dot{m} is the mass flow rate of the collector liquid, lb/hr,

c_p is the heat capacity of the collector liquid at constant pressure, Btu/(lb·°F),

T_1 and T_2 are the entrance and exit temperatures of the hot fluid, °F.

The rate of heat absorbed by the storage liquid is written as

$$Q_{HX} = (\dot{m}c_p)_s(t_2 - t_1), \text{ (Btu/hr)} \quad (7-2)$$

where

$(\dot{m}c_p)_s$ is the heat capacity flow rate of the storage fluid, Btu/(hr·°F)

t_1 and t_2 are the entrance and exit temperatures of the cold fluid, °F.

If an average temperature difference between the two temperature profiles, shown in Figure 7-12, could be determined, the heat flow rate from the collector fluid to the storage fluid could also be written

$$Q_{HX} = (UA)_{HX}(\overline{\Delta T}) \quad (7-3)$$

where

$(UA)_{HX}$ is the heat conductance rate through the walls of the heat exchanger tubes and is a function of fluid flow rates and the tube material and surface area of the interior tubes, Btu/(hr·°F)

$\bar{\Delta T}$ is an average temperature difference which is most frequently expressed as a log-mean temperature difference, (LMTD), °F.

In Equation (7-3), if $\bar{\Delta T}$ (LMTD) can be determined, the size of the heat exchanger can be selected because the heat transfer surface area, A , can be calculated for given tube material and heat flow rate, Q_{HX} . The difficulty is that LMTD is not readily determinable because the temperature profiles along the heat exchanger are not fully known, and the exit temperatures of the hot and cold fluids are dependent upon the $(UA)_{HX}$ of the heat exchanger. The log-mean temperature difference is written as

$$LMTD = \frac{(T_1 - t_2) - (T_2 - t_1)}{\log_e \frac{(T_1 - t_2)}{(T_2 - t_1)}} \quad (7-4)$$

where the temperatures are indicated in Figure 7-12.

A method convenient for sizing heat exchangers, which is not dependent upon knowledge of T_2 and t_2 and only on T_1 and t_1 , utilizes a factor called heat exchanger effectiveness, ϵ_{HX} . Heat exchanger effectiveness is the ratio of actual rate of heat transfer to a theoretical maximum rate that would occur if the heat exchanger surface area were very large (infinite). Expressed in symbols,

$$\epsilon_{HX} = \frac{(\dot{m}c_p)_c(T_1 - T_2)}{(\dot{m}c_p)_{\min}(T_1 - t_1)} = \frac{(\dot{m}c_p)_s(t_2 - t_1)}{(\dot{m}c_p)_{\min}(T_1 - t_1)} = \frac{Q_{HX}}{Q_{TH}} \quad (7-5)$$

where

$(\dot{m}c_p)_{\min}$ is the smaller value of the heat-capacity flow rate for the collector or storage liquids. To simplify the symbolism, hereafter we will use C for the heat-capacity flow rate.

Q_{TH} is the theoretical maximum heat transfer rate.

Once ε_{HX} is known, the heat transfer rate is determined directly from Equation (7-5) rewritten as Equation (7-6)

$$Q_{HX} = \varepsilon_{HX} C_{\min} (T_1 - t_1) \quad (7-6)$$

The heat exchanger effectiveness can also be expressed in terms of the material used in the heat exchanger and fluid capacitance rates as

$$\varepsilon_{HX} = \frac{1 - \exp\left[-\frac{UA}{C_{\min}} \left(1 - \frac{C_{\min}}{C_{\max}}\right)\right]}{1 - \frac{C_{\min}}{C_{\max}} \exp\left[-\frac{UA}{C_{\min}} \left(1 - \frac{C_{\min}}{C_{\max}}\right)\right]} \quad (7-7)$$

in which UA/C_{\min} is designated the number of transfer units, abbreviated NTU. The equation is presented graphically in Figure 7-14 and is valid in general for a counterflow heat exchanger. A parallel flow heat exchanger would have a different expression for ε_{HX} . The number of transfer units, NTU, is related to heat exchanger surface area and heat transfer coefficient, U , as

$$NTU = \frac{UA}{C_{\min}} \quad (7-8)$$

Typical values of the overall heat transfer coefficient, U , for shell-and-tube exchangers made of metal vary from 80 to 105 Btu/(hr·ft²·°F), depending upon fluid temperature and types of fluid used. Low values are appropriate for low flow rates, and high values of U for high flow rates. For different fluids, flow rates and heat exchanger sizes, ε_{HX} for counterflow heat exchangers can be determined from Figure 7-14.

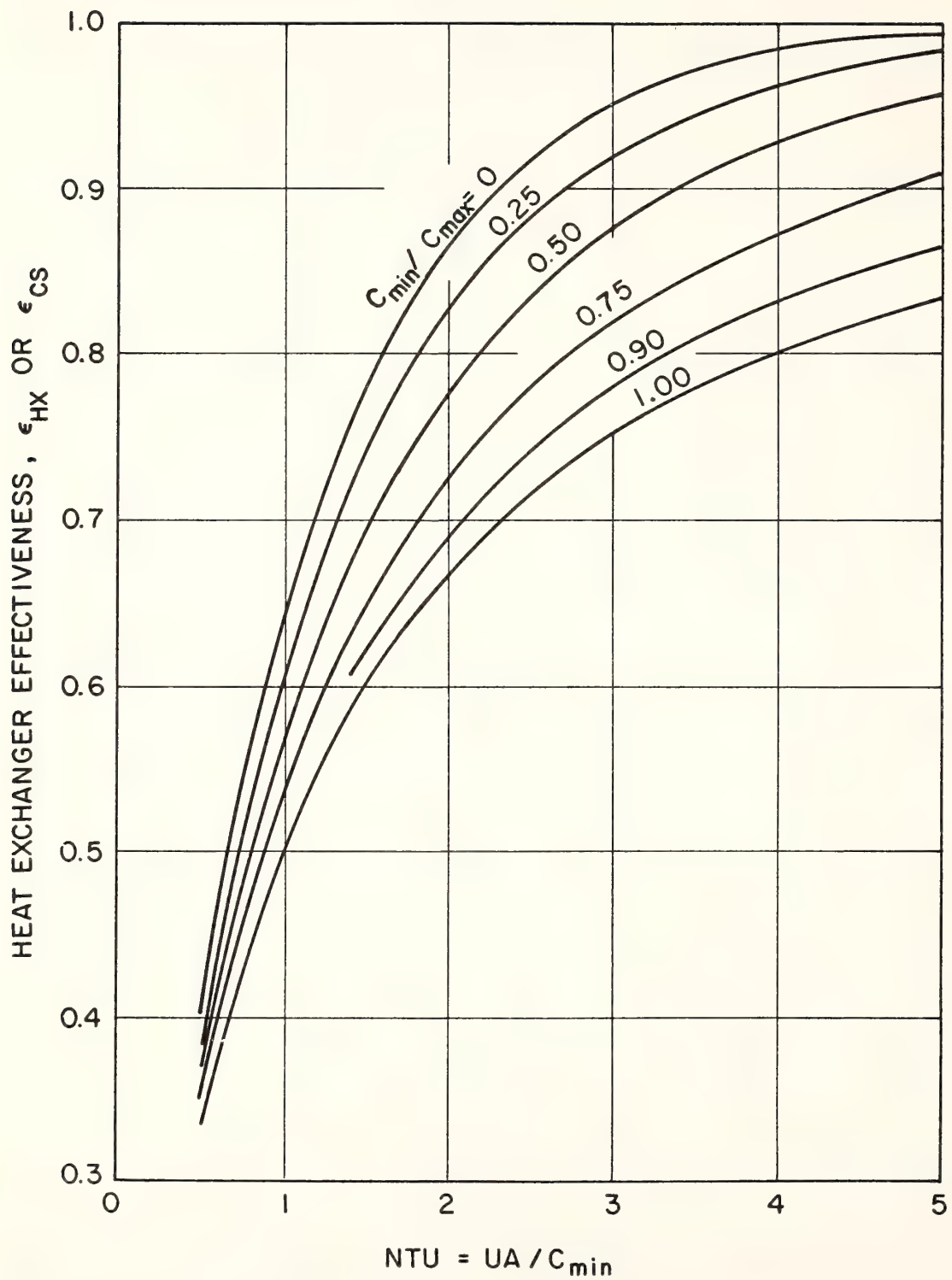


Figure 7-14. Effectiveness for a Counterflow Single-Pass Shell-and-Tube Heat Exchanger

Example 7-1 - Select a heat exchanger for a liquid system with a collector area of 500 ft². The collector has the following characteristic parameters:

$$F_R(\tau\alpha)_n = 0.7$$

$$F_R U_L = 0.93 \text{ Btu}/(\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F})$$

Assume a particular time of day when the intensity of solar radiation on the tilted collector, I_T , is at a near-maximum level of 300 Btu/(hr·ft²), inlet fluid temperature, to the collector, T_i , is 130°F, storage water temperature is 128°F, and ambient temperature, T_a , is 30°F. The collector fluid is a 50 percent mixture of ethylene glycol.

Solution - The heat delivery rate from the collector is

$$Q_c = A_c [I_T F_R(\tau\alpha)_n - F_R U_L (T_i - T_a)]$$

$$Q_c = 500 [300(0.7) - 0.93(130 - 30)]$$

$$Q_c = 58,500 \text{ Btu/hr}$$

To size the heat exchanger in the collector loop, flow rates for the collector and storage liquids are needed. The desired flow rate through the collector is 0.02 gpm/ft² of collector, so the total flow is 500 × 0.02 = 10 gpm. The rate of storage water flow through the heat exchanger is typically twice the rate through the collector, or 20 gpm.

The heat capacitance flow rate for the collector fluid is calculated from $C_c = (\dot{m}c_p)_c$. The temperature rise through the collector is determined as $T_o = T_i + Q_c/C_c$. From Figure 4-9, $c_p = 0.8 \text{ Btu}/(\text{lb}\cdot^\circ\text{F})$

and from Figure 4-10, the density is 1.048 g/ml which is equal to 8.34 x 1.048 = 8.74 lb/gal. Thus

$$C_c = 10\left(\frac{\text{gal}}{\text{min}}\right) \times 8.74\left(\frac{\text{lb}}{\text{gal}}\right) \times 60\left(\frac{\text{min}}{\text{hr}}\right) \times 0.8 \frac{\text{Btu}}{\text{lb}\cdot^{\circ}\text{F}}$$

$$C_c = 4195 \text{ Btu/hr}\cdot^{\circ}\text{F}$$

and

$$T_o = 130 + 58,500/4195 = 144^{\circ}\text{F}.$$

The heat capacitance rate of the storage water at a temperature of about 130°F is,

$$C_s = 20\left(\frac{\text{gal}}{\text{min}}\right) \times 8.34\left(\frac{\text{lb}}{\text{gal}}\right) \times 0.985^{\dagger} \times 60\left(\frac{\text{min}}{\text{hr}}\right) \times 1\left(\frac{\text{Btu}}{\text{lb}\cdot^{\circ}\text{F}}\right) = 9858 \text{ Btu/hr}\cdot^{\circ}\text{F}$$

Clearly, $C_c < C_s$, and $C_{\min}/C_{\max} = 4195/9858 = 0.43$. To choose a heat exchanger, calculate ε_{HX} ,

$$\varepsilon_{\text{HX}} = \frac{58,500}{4195(144-125)} = 0.87.$$

From Figure 7-14, select NTU for $\varepsilon_{\text{CS}} = 0.87$, $C_{\min}/C_{\max} = 0.43$, NTU = 2.8. The area of heat exchanger tube surface required, for an estimated $U = 120 \text{ Btu}/(\text{hr}\cdot\text{ft}^2\cdot^{\circ}\text{F})$, is

$$\frac{AU}{C_{\min}} - \text{NUT} = 2.8$$

$$A = \frac{2.8(4195)}{120} = 97.9 \text{ ft}^2$$

With this information a heat exchanger can be selected from a manufacturer's catalog.

[†]0.985 is the specific gravity of water at 130°F (See Figure 4-10)

Installation

Heat exchangers should be installed as close to the storage tank as possible. Pressure drops in heat exchangers are relatively large, and short pipe lengths in the fluid loops will minimize overall pressure drops and requirements for pumping power, and also will reduce heat losses.

Supports for heat exchangers are generally necessary because of size and weight. The entire outside surface of the heat exchanger should be well insulated, including the shell and the support frame. Before insulating, however, the pipe connections should be tested for leaks.

LOAD HEAT EXCHANGER

There are several types of heat exchangers that may be used to transfer heat from storage to the rooms. Solar heated water may be circulated through radiant panels, through fan-coil units, through baseboard heating strips, or through duct coils of central air heating systems. If water temperature to a radiant wall, floor, or ceiling panel is 100°F to 120°F and a large panel area is used, room heating will be adequate. For fan-coil units and duct heating coils, water temperatures of 120° to 150° are needed. Baseboard heating strips require higher temperatures for effective operation, generally in the 160° to 190° range.

Load heat exchangers discussed in this module are water-to-air cross-flow types. As with liquid-to-liquid heat exchangers, water-to-air heat exchangers can be sized to deliver any desirable rate of heat delivery to the room. From practical considerations, the load heat

exchanger should be limited to a size such that water at about 145°F can meet the design heating load. The heat delivery rate from a water-to-air heat exchanger is determined by Equation (7-6) with ε_L substituted for ε_{HX} . If the heat delivery rate is equal to the heat loss rate from the building enclosure,

$$\frac{\varepsilon_L C_{\min}}{(UA)_L} = \frac{T_R - T_a}{T_S - T_R} \quad (7-9)$$

where

T_R is room temperature, °F

T_a is design outdoor temperature, °F,

T_S is storage water temperature, °F.

For a location where $T_a = 0^\circ\text{F}$, $T_R = 70^\circ\text{F}$ and $T_S = 140^\circ\text{F}$,

$$\frac{T_R - T_a}{T_S - T_R} = \frac{70 - 0}{140 - 70} = 1,$$

and when extremely cold temperatures are experienced outdoors, $T_R - T_a$ would be larger than 70°F and correspondingly the water supply temperature must be higher than 140°F to be able to provide adequate heat to the rooms.

If a large heat exchanger is used (large $\varepsilon_L C_{\min}$), water at lower temperature can supply the heating rate to meet the load. Let us assume that $\varepsilon_L C_{\min} / (UA)_L = 3$. In the example used above,

$$3(T_S - T_R) = T_R - T_a$$

$$3(T_S - 70) = 70 - 0$$

$$T_S = 93^\circ\text{F}$$

Storage water temperature at 93°F through the heat exchanger would deliver heat at a rate sufficient to heat the room air. The load heat exchanger effectiveness, and the size (UA) of the heat exchanger, can be

determined with use of Figure 7-15. The example below illustrates the method.

Example 7-2 - Select a load heat exchanger such that when storage water temperature is 100°F, there will be sufficient heat transfer to meet a design building load. A heat loss calculation has been made for the building and $(UA)_L$ is 715 Btu/(hr·°F) with a design outdoor temperature of 0°F. Determine the design air flow and water flow rates through the heat exchanger.

Solution - To determine $\varepsilon_L C_{\min}/(UA)_L$, use Equation (7-9).

$$\frac{\varepsilon_L C_{\min}}{(UA)_L} (T_S - T_R) = T_R - T_a$$

$$\frac{\varepsilon_L C_{\min}}{(UA)_L} (100 - 70) = 70 - 0$$

$$\frac{\varepsilon_L C_{\min}}{(UA)_L} = \frac{70}{30} = 2.33$$

Select a heat exchanger effectiveness of 0.85. Then

$$C_{\min} = \frac{2.33 \times (715)}{0.85} = 1960 \frac{\text{Btu}}{\text{hr} \cdot ^\circ\text{F}}$$

Generally, for a water-to-air heat exchanger, C_{\min} is usually the heat capacitance rate for air. The required air flow rate is calculated as follows:

$$C_{\text{air}} = 60 \dot{V}_a \rho_a (c_p)_a \quad (7-10)$$

where

\dot{V}_a is the volumetric air flow rate, ft³/min

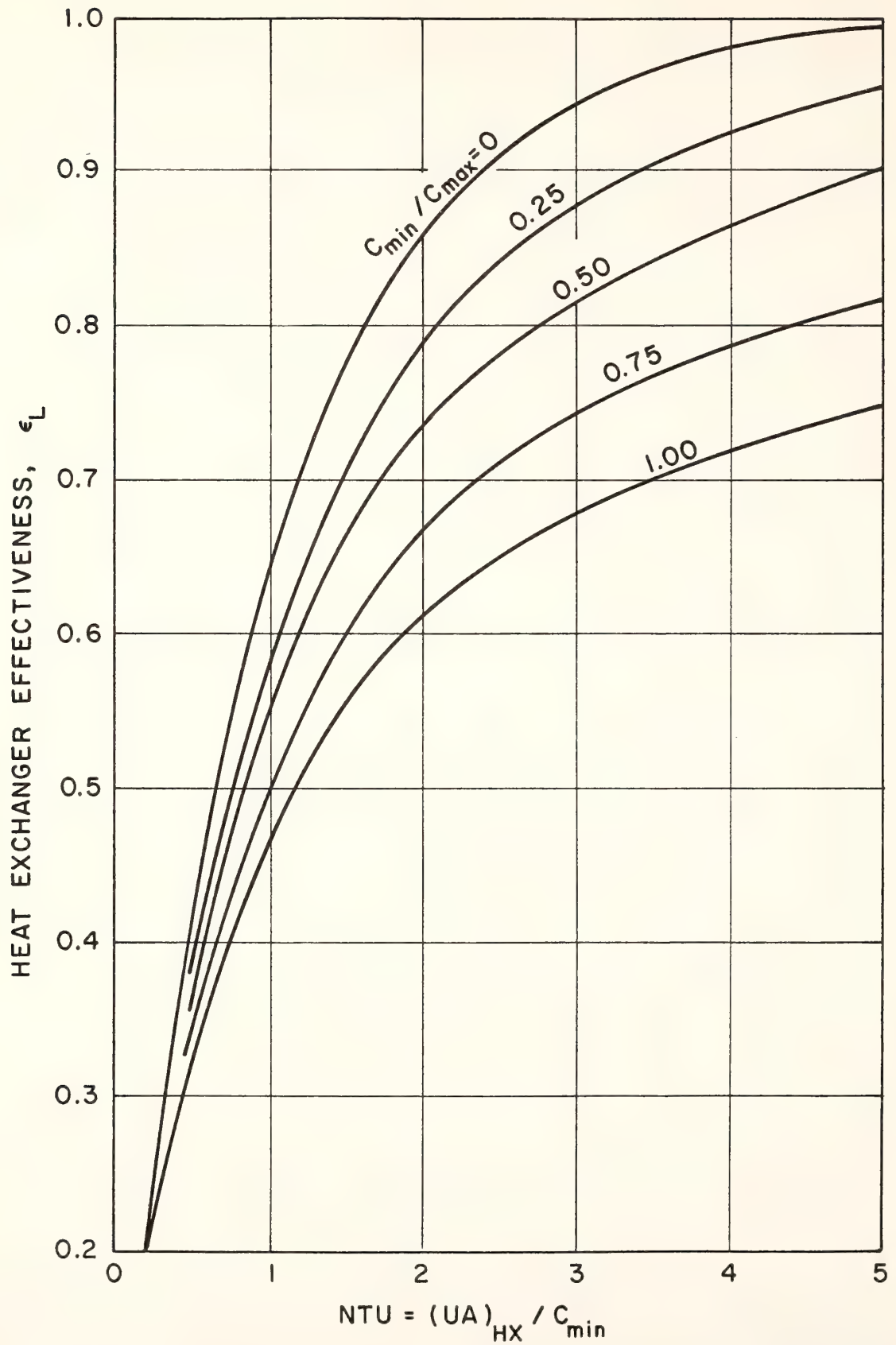


Figure 7-15. Effectiveness of Cross-Flow Water-to-Air Heat Exchanger

ρ_a is air density, lb/ft³

$(c_p)_a$ is specific heat of air.

At 70°F, $\rho_a = 0.075$ lb/ft³, $(c_p)_a = 0.24$ Btu/(lb·°F)

Thus,

$$\dot{V}_a = \frac{1960}{(0.075)(0.24)(60)} = 1815 \text{ cfm}$$

From Figure 7-15 select a heat exchanger with minimum surface area. Using the curve $C_{\min}/C_{\max} = 0.25$, NTU = 2.6, the overall heat transfer coefficient, U, can vary from 10 to 30 Btu/(hr·ft²·°F) for water-to-air heat exchangers. In this example use U = 25. Thus a heat exchanger with a surface area, A, equal to

$$A = \frac{(2.6)(1960)}{25} = 204 \text{ ft}^2$$

is required.

The water flow rate through the heat exchanger is calculated below:

$$C_{\text{water}} = \frac{1960}{0.25} = 7840 \frac{\text{Btu}}{\text{hr} \cdot ^\circ\text{F}},$$

and because $C_{\text{water}} = 60G_w\rho_w \cdot (c_p)_w \cdot S_w$

where

G_w is flow rate, gal/min,

ρ_w is water density, (8.34) lb/gal

$(c_p)_w$ is specific heat of water, (0.993) Btu/(lb·°F)

S_w is specific gravity of water at a given temperature (1).

$$G_w = \frac{7840}{(8.34)(1)(0.993)(60)} = 15.8 \text{ gpm.}$$

It should be noted that the required air flow rate is large for a building with a heat loss rate of 715 Btu/(hr·°F), and the heat

exchanger size is large. If the design condition is selected for storage water temperature at 140°F, a smaller air flow rate and heat exchanger size would result, but with the storage temperature below 140°F, the heat flow rate from solar storage would be unable to meet the design load, so auxiliary heating would be required.

DOUBLE-WALLED HEAT EXCHANGER

To prevent contamination of household water and water mains, double-walled heat exchangers should be used for transfer of heat from solar storage to domestic water. Double-walled heat exchangers are not as commonly available as single-walled shell-and-tube heat exchangers. Liquid leaks through either wall must be readily detectable so that the unit may be repaired or replaced. One type utilizes a tube wrapped around the outside of the preheat tank. A Roll-Bond[®] sheet is particularly suitable for this purpose. Other designs involve concentric serpentine tubes with water inside the inner tube and outside the outer tube. The annulus between the tubes is vented so that leaks can be detected. Another design involves parallel tubes cross-connected with many heat transfer fins to conduct heat from the hot tube to the cold tube, similar to baseboard heating strips.

A design procedure for double-walled heat exchangers is not well established but fortunately, sizing is not critical for domestic water pre-heating. Typical flow rates for both sides of the heat exchanger are 2 to 3 gpm. Temperature drops of 5°F to 10°F across the heat exchanger are satisfactory and temperature difference of about 20°F between the hot and cold fluids is tolerable. Since the water circulation rate is small, pressure drop is low and only fractional horsepower circulation pumps (1/10 to 1/30) are needed.

PUMPS

Centrifugal pumps are best suited for liquid-heating solar systems. Typical performance curves for a centrifugal pump are shown in Figure 7-16.

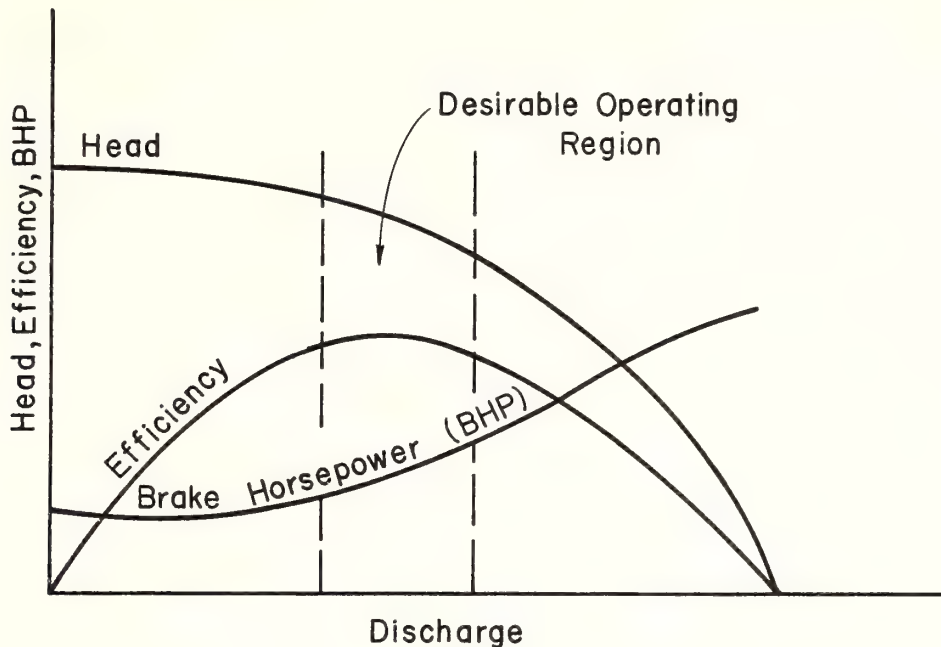


Figure 7-16. Typical Performance Curves for a Centrifugal Pump

The "head" is the pressure generated by the pump expressed in feet of water and is the difference in pressure between the suction and discharge sides. The head generated by the pump is equal to the sum of the pressure drops across the components of the circulation loop. The efficiency of the pump is the ratio of fluid (or brake) power generated by the pump, to the electrical power delivered to the motor. From the curves in Figure 7-16 it is readily seen that pumps should be chosen to circulate the desired flow rate at required head to operate near peak efficiency. The size of motor needed for the pump is determined by the brake horsepower and pump efficiency.

Since the head produced by a pump is exactly balanced by the head losses in the system, it is necessary to estimate the head losses in the piping loops, across collectors, and through heat exchangers before selecting pumps. A chart for estimating friction losses in copper tubing is shown in Figure 7-17. Similar charts are available for other types of piping from suppliers. A nomograph for estimating head losses across valves and fittings is shown in Figure 7-18. Head losses for collectors and heat exchangers are given in manufacturers' catalogs and brochures.

To estimate head losses in a particular loop, a piping layout (schematic) should be made with valves, fittings, filters, and other components shown in the loop. Pipe sizes should be selected so that flow velocities will not exceed about 7 ft/sec. With discharges selected and pressure losses calculated for each circulation loop, pump selections can be made from manufacturers' catalogs.

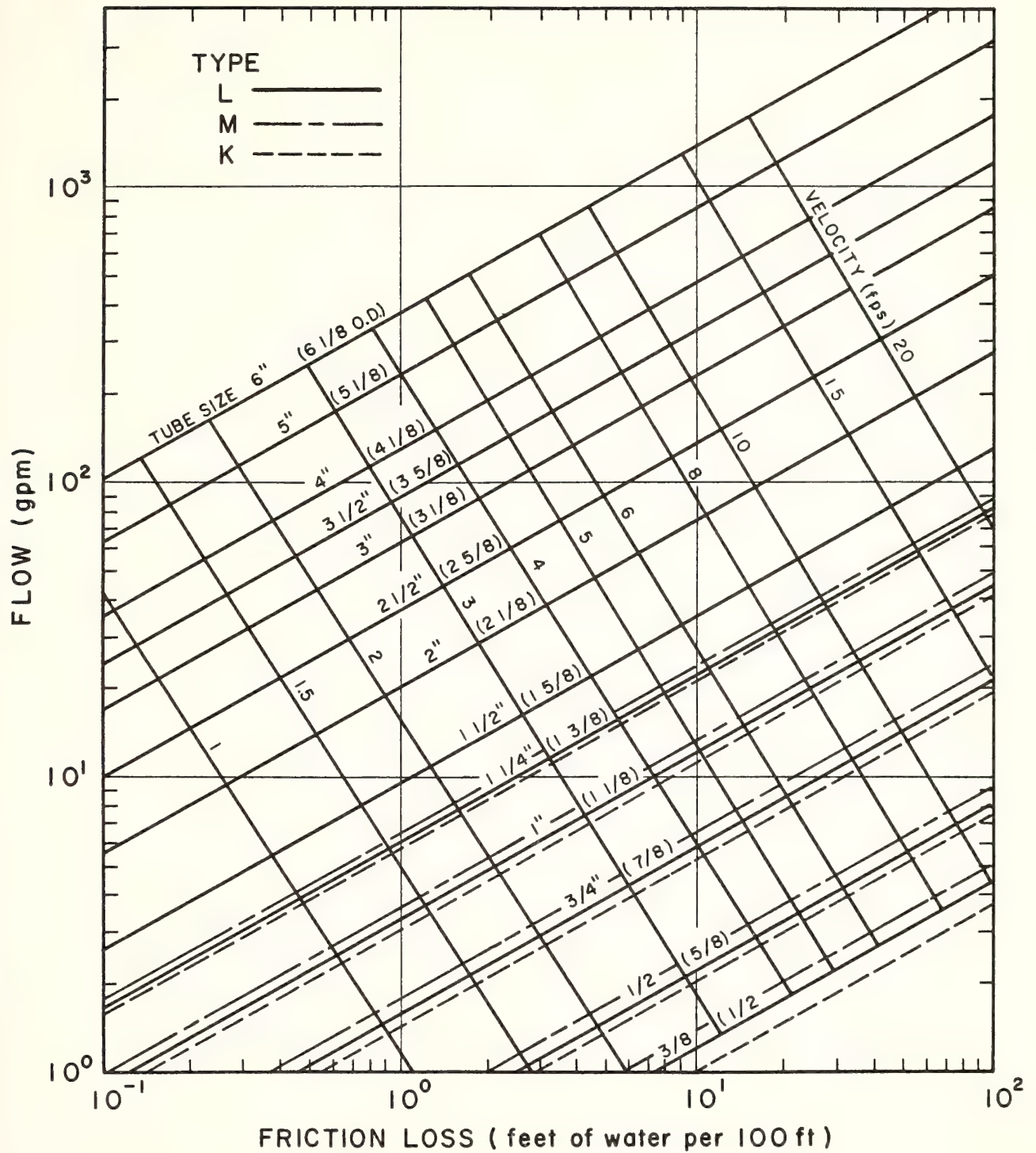


Figure 7-17. Friction Loss in Copper Tubing

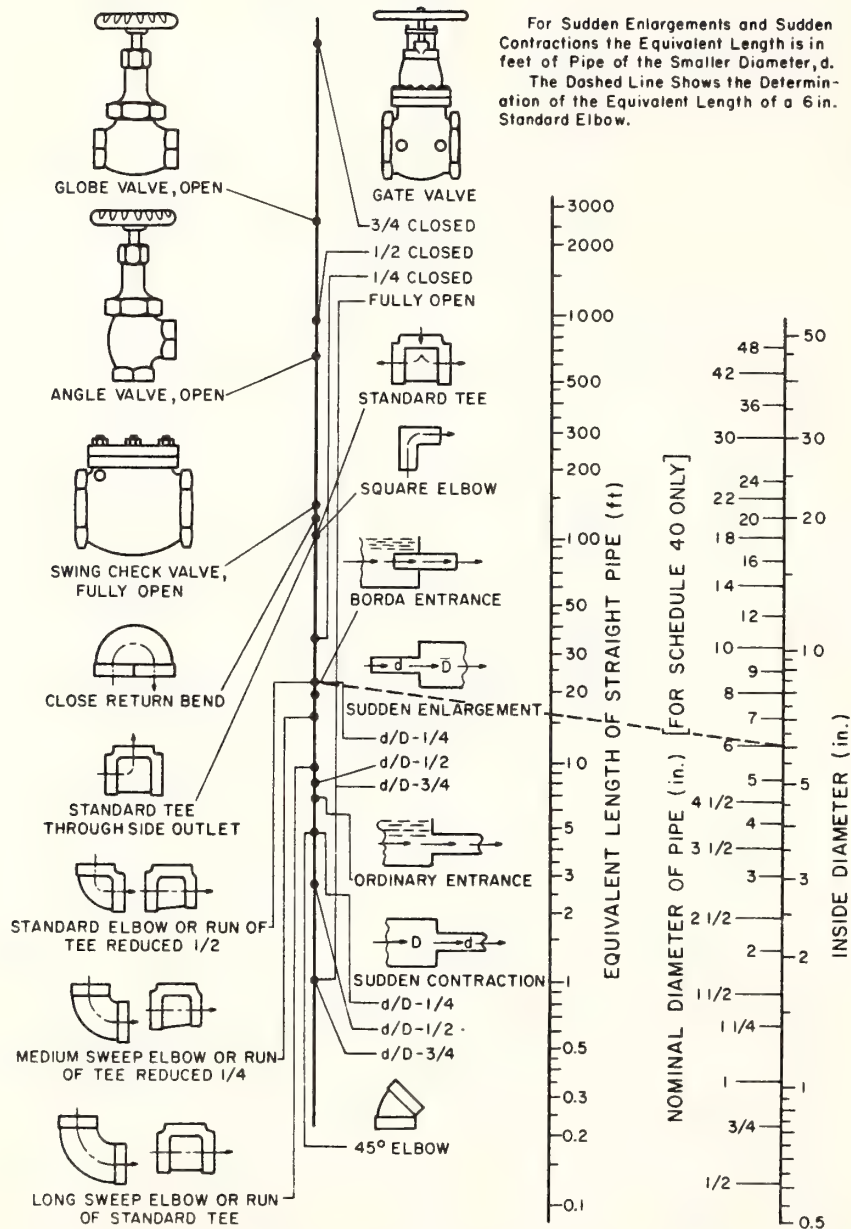


Figure 7-18. Pressure Loss in Various Elements

REFERENCES

1. Dickinson, W.S., Neifert, R.D., Löf, G.O.G., and Winn, C.B., (1975). "Performance Handbook for Solar Heating Systems". Presented at the 1975 International Solar Energy Society Congress, University of California at Los Angeles, California.
2. Klein, S.A., Beckman, W.A., and Duffie, J.A., (1975). "A Design Procedure for Solar Heating Systems". Presented at the 1975 International Solar Energy Society Congress, University of California at Los Angeles, California.
3. Duffie, J.A. and Beckman, W.A., (1974). Solar Energy Thermal Processes, John Wiley and Sons, New York.
4. Pickering, Ellis E., (1975). "Residential Hot Water Solar Energy Storage", Proceedings of the Workshop on Solar Energy Storage Subsystems for the Heating and Cooling of Buildings, Charlottesville, Virginia, April 1975.
5. Peltzman, E.C., "Differential Thermostats for Solar Energy Systems", Rho Sigma, Inc., 15150 Raymer Street, Van Nuys, CA 91405.
6. Krieth, F., (1966). Principles of Heat Transfer, International Textbook Co., Scranton, PA.
7. ASHRAE (1972) Handbook of Fundamentals, ASHRAE, New York.
8. ASHRAE (1972) Guide and Data Book, ASHRAE, New York.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 8

COMPONENTS OF AIR SYSTEMS

HEAT STORAGE

CONTROLS

HEAT EXCHANGERS

BLOWERS, AIR HANDLERS

SOLAR ENERGY APPLICATIONS LABORATORY
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OBJECTIVE

The objective is to present information on components of air systems so that participants will be able to:

1. Select an appropriate-size heat storage unit.
2. Select blowers.
3. Install components properly to operate air systems.

INTRODUCTION

Air systems contain the same types of components as liquid systems, namely collectors, thermal energy storage units, heat exchangers for domestic water heating, prime movers to circulate heated fluid streams, and automatic controls. Module 7 contains general discussion of system components as well as comments on specific items applicable for liquid systems. This module is limited to discussion of specific components of air systems.

HEAT STORAGE

PEBBLE BEDS

Rock pebbles are commonly used in air-heating solar systems to store sensible heat. Hot air from the collectors flows through a bed of pebbles and heat is rapidly transferred from the air to the rocks. The air cools as it passes through the pebble bed and leaves at a temperature nearly equal to the temperature of the rocks at the exit end of

storage. The resulting temperature stratification in the bed is an advantage to collector operation and also to the supply of heat from storage to the rooms. For collection, cool air is returned to the collectors and operation is always at highest efficiency. When stored heat is delivered to the rooms, air flow direction through storage is reversed so that cool room air enters the cool end of the pebble bed, is heated as it flows through warm layers of rocks, and leaves at the opposite (hot) end of storage. Typical variations of temperatures during a full cycle of charging and discharging of heat in a pebble bed are shown in Figure 8-1.

Although it is not essential, heated air from collectors should enter at the top of the pebble bed in vertical-flow storage beds and cool room air should be circulated from the bottom toward the top. In this arrangement the bottom of storage, which is coolest, is adjacent to the floor, so heat loss to the ground is minimized.

Pebble-Bed Containers

Pebble-bed containers can be wood frame boxes, poured concrete bins, masonry (concrete block) structures, or steel bins. Wood frame boxes are easily constructed even in places where access is limited. All containers must be structurally adequate and joints must be prevented from cracking either from the force of pebbles inside the container or by settlement of the foundation.

Air leaks from containers should be minimized. A wooden box container is illustrated in Figure 8-2, with plenums at the top and bottom. The bottom plenum is created by supporting the pebbles on a screen resting on steel frames or on spaced concrete blocks.

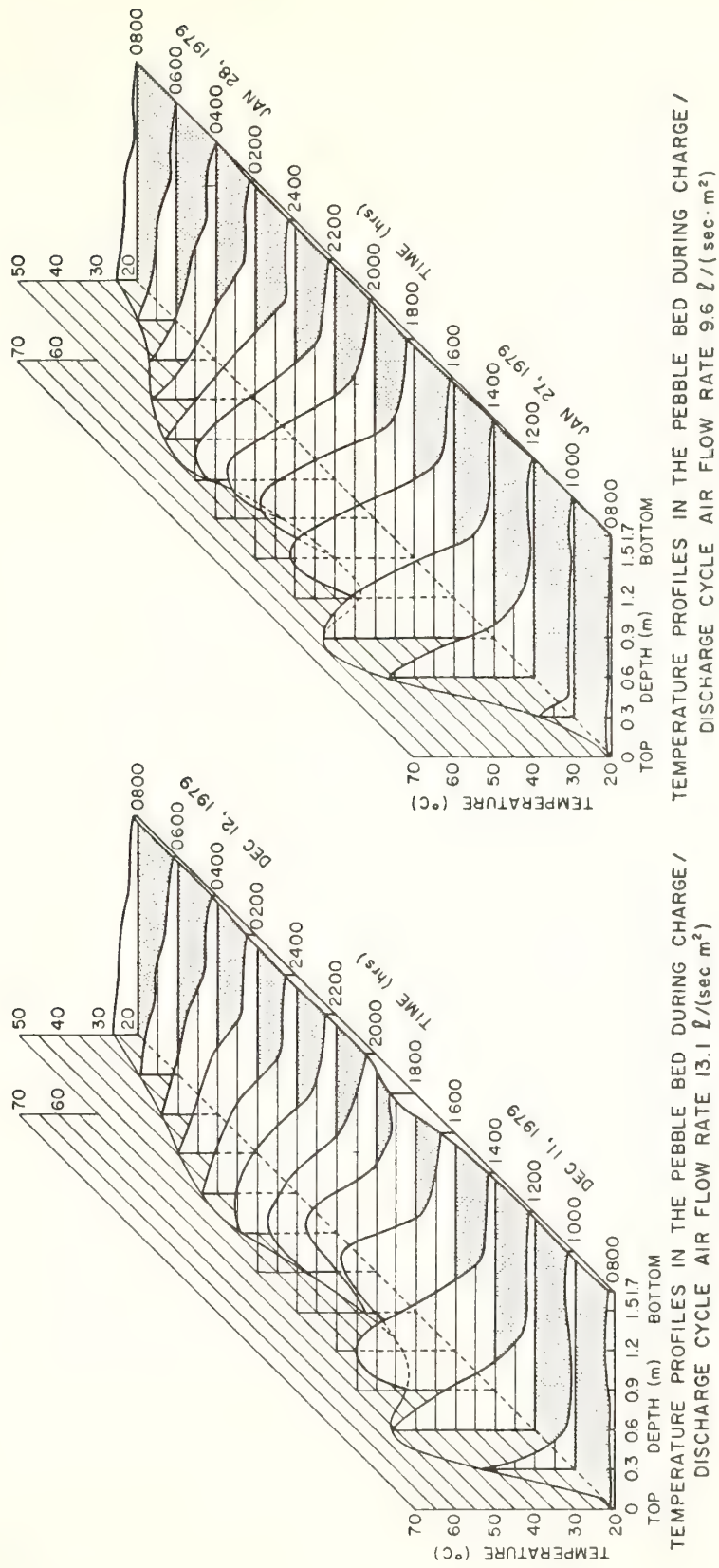


Figure 8-1. Typical Temperature Stratification in a Pebble Bed

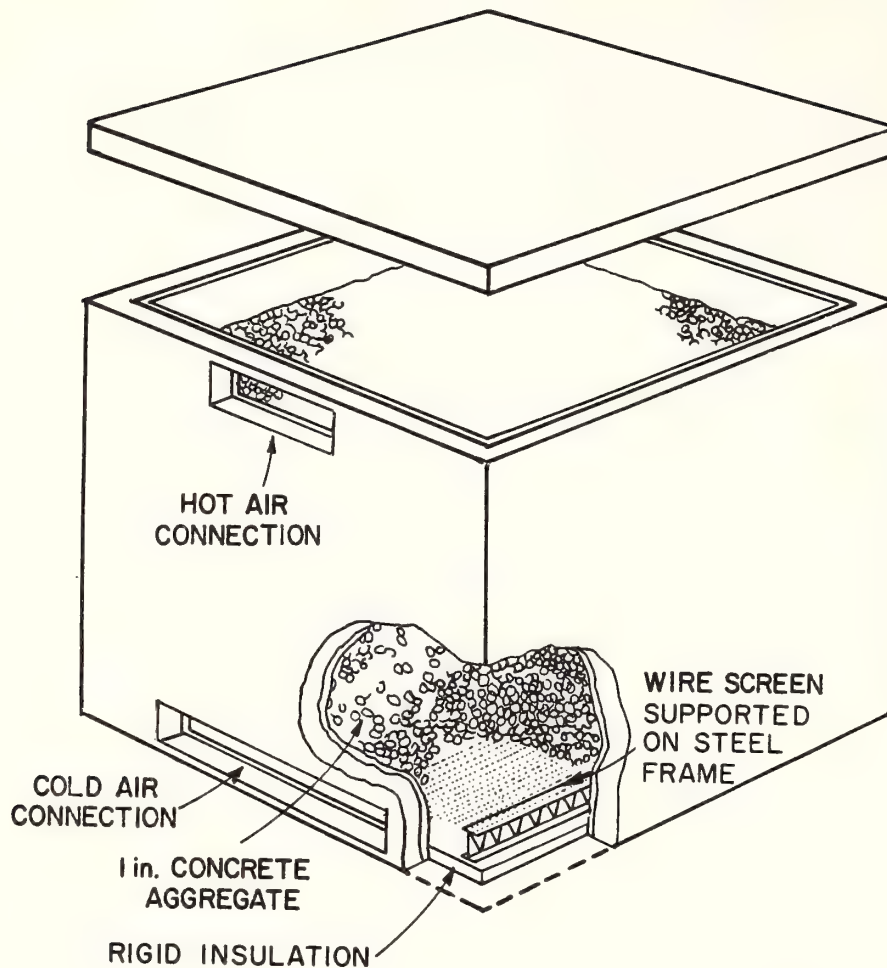


Figure 8-2. Pebble-Bed Heat Storage Unit

Concrete blocks or other types of masonry may be used for the walls of pebble beds. They should be reinforced and insulated, and the interior should be lined or sealed to prevent air leakage. Insulating the walls to achieve R-10 to R-13 rating is generally satisfactory.

A container made from poured concrete walls is relatively economical when constructed at the same time as basement walls. Only two additional walls in one corner of the basement are needed to form

the rock bin. Insulation should be placed against the inside walls, and the construction should be essentially air-tight.

Sizing the Pebble Bed

A maximum depth of about 8 feet of pebbles is recommended to limit pressure drops through the bed. The cross-sectional area of the container (perpendicular to flow direction) should be sized to achieve a superficial flow velocity of about 20 ft per minute. In 5 to 8 ft of path length, the pressure drop will be about 0.10- to 0.16-inch water gauge through a pebble bed containing 3/4- to 1.5-in diameter pebbles. A curve of pressure drop through a pebble bed as a function of superficial air velocity through the bed is shown in Figure 8-3.

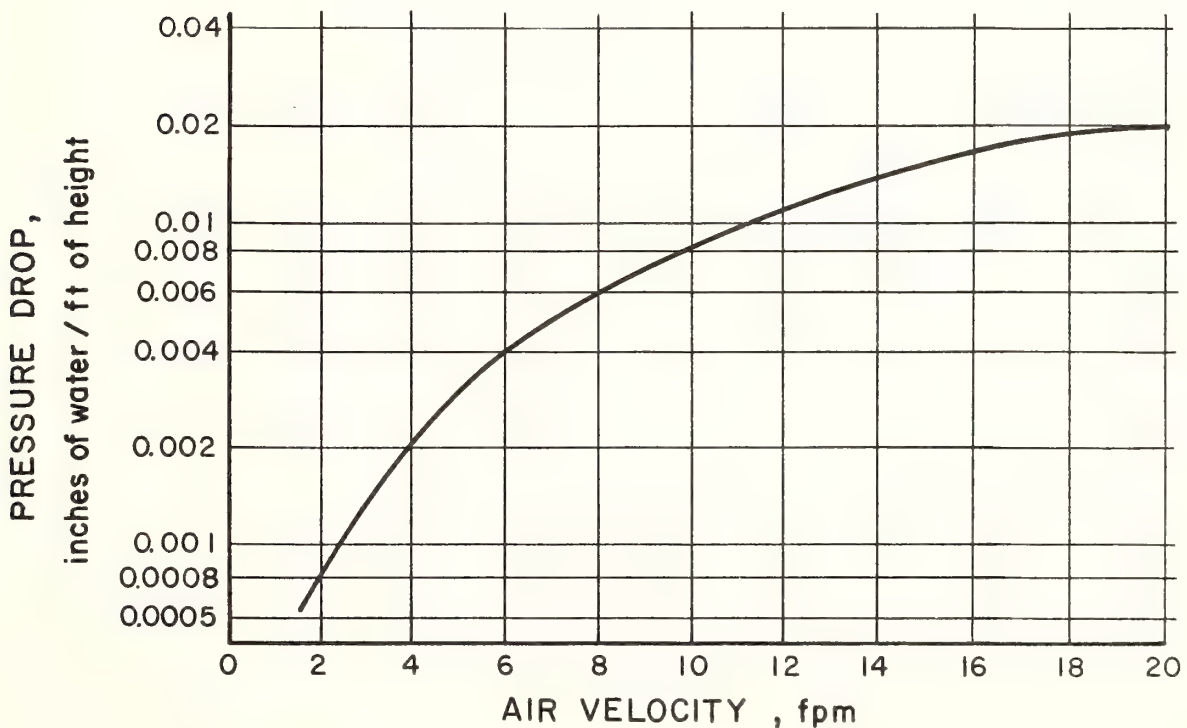


Figure 8-3. Pressure Drop Through a Pebble Bed with 3/4- to 1-1/2-Inch Rocks

The effect of pebble size and air velocity on the pressure loss through the bed is indicated in Table 8-1. At a typical face velocity of 20 ft/min, the pressure loss in a pebble bed 6 feet deep is 0.17 inch water gauge if uniform 3/4-inch rock is used, and 0.09 inch water gauge if 1½-inch rock is used. If the rock is a mixture of sizes between these limits, the pressure loss is approximately the same as that prevailing with the smallest size, i.e., the 3/4-inch material.

Table 8-1
Pressure Loss in Pebble Beds

Air Face Velocity Feet Per Minute	Pressure Loss Inches Water Gauge/foot of length	
	3/4-Inch Pebbles	1½-Inch Pebbles
10	0.008	0.0025
15	0.017	0.008
20	0.028	0.015
25	0.046	0.023

The volume of pebble bed recommended for air systems is one-half to one cubic foot for each square foot of collector area. Thus, for 400 ft² of collectors, 200 to 400 ft³ of rocks are desired. With 200 ft³ the container should have a cross-sectional area of about 40 ft², and the depth will be about 5 ft. For 400 ft³, the cross-sectional area should be about 50 ft², and the depth will be about 8 ft.

Horizontal Pebble Beds

Pebble beds with horizontal flow can be constructed if vertical space is limited. Channeling of airflow across the top of the bed is the principal concern because of settlement of the rocks in the container. Barriers along the top placed perpendicular to the flow will reduce channelling, but air flow along the top could still persist if there is a gap. To maintain uniform flow and temperature stratification, horizontal separators, such as plastic films or sheets, placed about 12 inches apart as shown in Figure 8-4 can be helpful. It is more difficult, because of space restrictions, to maintain large cross-sectional areas and short path lengths in horizontal pebble beds. Superficial flow velocities greater than 20 ft/min and path lengths longer than 10 ft, both of which may be necessary, will add substantially to electrical power requirements for air circulation.

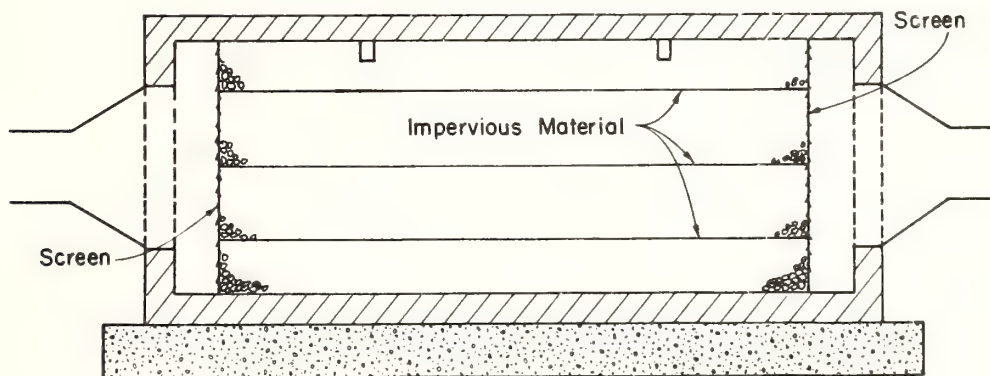


Figure 8-4. Horizontal-Flow Pebble Bed

Rocks for the Pebble Bed

Rocks suitable for use as concrete aggregate are suitable for pebble-bed storage. Uniformity in size of pebbles is important to create uniform airflow through the pebble bed. Crushed rock or river

gravel should be screened for sizes usually from 0.75 to 1.5 inch and washed before placement if not reasonably clean and free of debris. When filling, the rock should be placed in a manner which minimizes fracturing. Small particles which fill the spaces between the pebbles will restrict air passage and cause non-uniformity of air flow and heat storage within the pebble bed.

PHASE-CHANGE STORAGE

Some phase-change storage materials (PCM's) for use with air heating systems are commercially available. Salt hydrates are packaged in tubular or capsule form, and heat is transferred to and from them by blowing air across the encapsulated materials. A large amount of heat can theoretically be stored in a small volume of material, but presently available compounds have freezing and melting temperatures that are too low (80°F to 95°F) for conventional warm-air heating systems. To store heat at higher temperatures, the temperature of the melted material must be increased, but because volume is limited, large quantities of sensible heat cannot be stored at higher temperatures. PCM costs are high, and temperatures are not suitable for forced air heating systems. Through continued research, appropriate PCM devices for space heating systems may become practical in the future.

SYSTEM CONTROLS

Components of a basic control subsystem for an air-heating solar system consist of sensors, differential thermostats, and relays or switches to activate the mechanical devices. The components are

substantially the same as for liquid-heating solar systems explained in Module 7.

PRINCIPLES OF OPERATION

Collecting Solar Heat

A schematic diagram of a one-blower air-heating solar system is shown in Figure 8-5. The system consists of flat-plate collectors, pebble-bed storage, air handler, auxiliary furnace, and a domestic water-heating subsystem. Depending on the mode of operation, the dampers in the air handler open and close to circulate air through appropriate ducts.

There are three temperature sensors: S1 located near the top of the collector, S2 located at the bottom of storage or in the air duct supplying the collector, and S4 near the bottom of the domestic water pre-heat tank. The room thermostat is designated S3 and consists of two-stage contacts for heating. Sensor S1 may be placed either in the air stream near the top of the collector or attached to the back side of the absorber plate. Temperature difference settings will be different for the two placements, with a larger difference setting required if the sensor is attached to the absorber plate. Normally S1 is placed in the air stream within the collector.

To initiate collection of solar heat, the temperature difference between S1 and S2 is considered. If the difference is greater than a preset amount, say 15°F to 20°F, the blower in the air handler is activated and heat is delivered from the collectors. Unlike a liquid system, the air system can be controlled to deliver heat directly to the rooms as well as to storage.

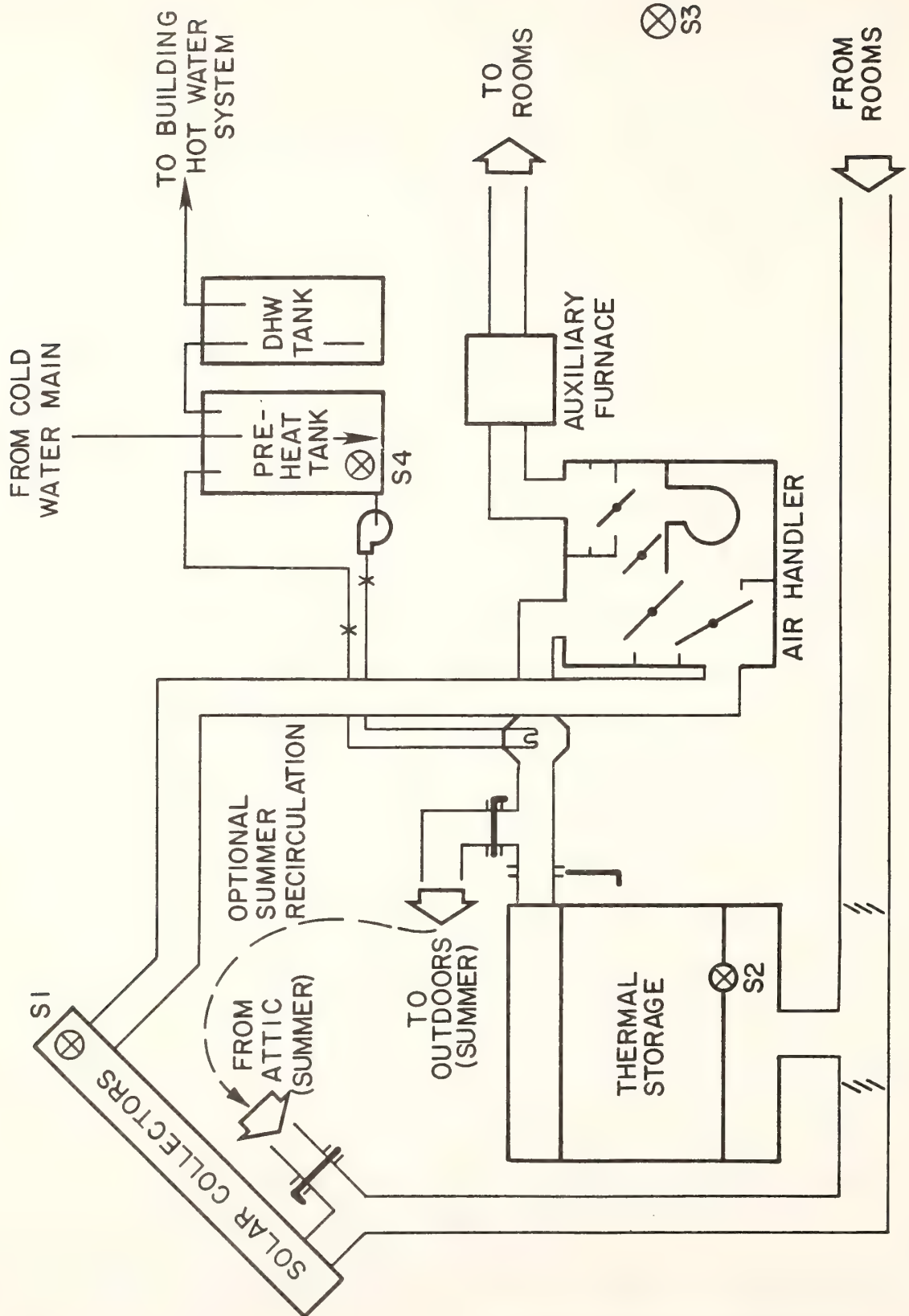


Figure 8-5. Sensor Locations in Typical Air-Heating System

Delivering Heat to Storage - Simultaneously with blower activation, dampers in the air handler are automatically positioned to circulate air through the blower and to the top of storage. The motorized damper at the bottom of storage is closed and air is directed to the collectors. Since all of the useful thermal energy collected (above room temperature) is stored in the rock bed, the air temperature at the bottom of storage is cool. The pebbles at the bottom of storage will be at room temperature at the start of a collection day and S2 will remain constant until the thermal front advances all the way through the pebble bed. With proper sizing of storage, S2 will typically remain at room temperature through the entire collection day during the winter as indicated in Figure 8-1.

Temperature variations at sensors S1 and S2 are shown for a typical winter day in Figure 8-6. The blower is activated at point ① when the temperature difference is, for example, about 15°F. As soon as the blower starts, the collector is cooled and air temperature at S1 drops to point ②. If the difference is less than the minimum set point, the blower stops and restarts a few minutes later. A minimum temperature difference setting for shutting off the blower may be as low as 2°F or 3°F. As the intensity of solar energy increases during the day, air temperature at S1 increases and reaches a maximum at midday. In the afternoon, air temperature at S1 continually decreases because of declining solar intensity, and the blower stops at point ③ when the temperature difference reaches the minimum set point. The collector temperature then rises slightly (to point ④), after the blower stops because of additional solar radiation on the absorber and heat retention in the collector. If point ④ is high enough, the blower may restart and run for a few minutes before again shutting down.

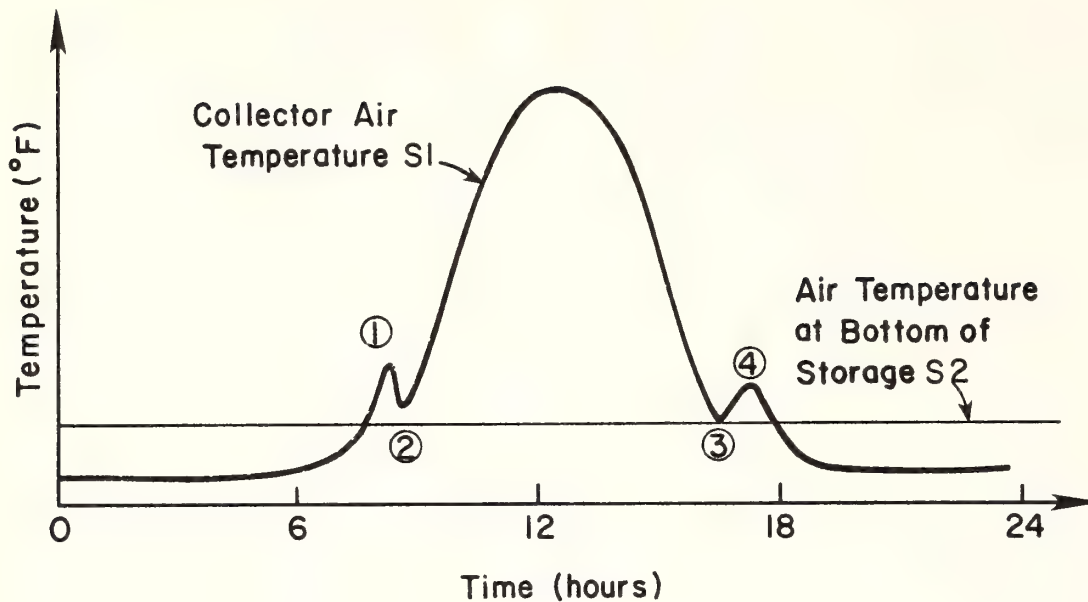


Figure 8-6. Typical Temperature Variations at S1 and S2

Normally, with a start set point of 15°F difference between S1 and S2 and a stop set point of 2°F to 3°F, collection will begin on sunny winter days between 0800 and 0830 hrs, and stop between 1630 and 1700 hrs in the afternoon. With a start-to-stop temperature set point ratio as high as 5 to 7, on-and-off cycling of the blower will not usually occur either at the start of collection or at termination of collection.

The foregoing temperature difference settings are small enough to provide virtually the maximum collectable solar heat to storage and to direct use. When air is supplied to the collector at about 70°F, these settings permit heat delivery to storage and directly to the rooms at temperatures as low as 85° (start-up) and even at 72° (shut-down). Because these temperatures may be too low for comfortable supply to the rooms, differential settings may, in common practice, be considerably higher. In a widely used air system, for example, the manufacturer

recommends (and provides factory settings) a start-up temperature difference of 40°F and a shut-down difference of 25°F. These settings assure air deliveries to the heated space at temperatures no less than 95°F.

Delivering Heat to Rooms - When solar heat is being collected and thermostat S3 calls for room heating, dampers in the air handler and at the bottom of storage open and close appropriately to direct the air stream to the rooms. If the temperature of the air stream from the collector is not sufficiently warm to heat the room adequately, the second stage contact of the thermostat is completed and the auxiliary furnace is activated. Since the auxiliary furnace is sized to heat the rooms adequately by itself, room temperature will soon rise above the thermostat setting, the auxiliary furnace will be deactivated, dampers in the air handler will be repositioned, and solar heated air will then be directed to storage.

Heating from Storage

Only the thermostat S3 is required to control room heating from storage. The first stage calls for solar heat from storage. When heat delivery is inadequate to meet the room requirements, the second stage of the thermostat activates the auxiliary furnace. Room air is always circulated through storage whether auxiliary heating is required or not; thus all stored heat, regardless of its temperature, is utilized for space heating.

Domestic Water Pre-heating

The control for pre-heating domestic water is a differential thermostat for sensors S1 and S4. The temperature difference setting to start pre-heating may be about 15°F, and to stop pre-heating, the set point may be as low as 3°F. There is considerable variation in practice, however, settings as high as 40° "on" and 25° "off" being observed.

For water heating during the summer a by-pass duct may be used with a summer-winter damper that permits air circulation only through collector and water coil. A temperature limit switch may be advisable to prevent boiling. With the system shown in Figure 8-5, where attic air is drawn through the collector and hot air is discharged outdoors, it is not necessary to include a temperature limiter. However, if hot air is recirculated through the collector during the summer, air temperatures may become high enough for water to boil in the pre-heater tank. This condition is usually avoided by using S4 to turn off the blower and water pump when a preset temperature limit is reached.

TEMPERATURE SENSORS

Thermistors are usually employed as temperature sensors in air systems because they deliver ample voltage (2 to 6 volts) to the controller, and the signal is nearly linear. Sensors with output voltage that is linear with temperature provide more consistent control for solar energy collection and water heating because the temperature difference setting is independent of absolute temperatures. That is, with "linear output" sensors, temperature difference settings are more reliable.

INSTALLATION OF CONTROL HARDWARE

Control Panels

Control panels are usually compactly packaged for easy mounting. Connections to sensors and output devices are usually labeled and easy to attach. Some manufacturers include the control assembly with the air handler, and connections to dampers and the motor are wired at the factory. Only the sensor and thermostat connections are required for such units. For systems with damper motors outside the air handler, control wires will have to be attached separately. Instructions for installation should be provided with the system.

Location of Temperature Sensors

The sensor in the collector should be located near the exit of the top collector in an array with air flow from bottom to the top of the array. With or without forced air movement, the temperature is highest at the exit. The sensor at the bottom of storage should be in contact with the pebbles at the bottom. An alternative location for the sensor is in the bottom plenum or in the duct leading to the collector.

Locating a sensor S2 (see Figure 8-5) in the return duct to the collectors rather than at the bottom of storage can cause difficulty if positioned too close to the collector. Even with a damper between the sensor and the collector, cold air from the idle collector may settle downward in the duct and cool S2. Air circulation may thus be initiated even when storage is warmer than the air at the collector outlet. The system will operate until warm air from storage heats sensor S2 and periodic cycling can result. Convenience in installation may be served, however, by a duct-mounted sensor S2, so a location near the bottom of

the pebble bed, just beyond the junction with the duct returning cold air from the rooms, should then be used.

The sensor in the domestic water pre-heat tank should be located near the bottom where the coldest water is present.

Auxiliary Furnace Control

The auxiliary furnace is activated by the second stage of the room thermostat and shuts off when the thermostat set point is reached. To reduce room temperature fluctuation, the furnace is usually immediately activated if the temperature at the top of storage, determined by another sensor at that point, is below a preset level, such as 90°F. Use of such a control scheme will decrease slightly the solar contribution to the seasonal heating load but will increase comfort.

Control System Check-Out

To assure reliable performance, all operating modes of the system should be checked out after installation. To initiate heat collection, in the absence of sunshine, the terminals of sensor S1 at the controller can be short-circuited to simulate a high collector temperature. The blower should start, and if the room thermostat is at a low setting (below actual room temperature), air should flow through storage. The domestic water circulation pump should also start. While in this collection mode, increasing the room thermostat setting to the first stage contact should result in air circulation through the rooms. A further increase in thermostat setting, until the second stage is contacted, should activate the auxiliary furnace. With the short circuit removed (and little or no solar radiation), a thermostat setting slightly above

room temperature should start air circulation through storage to the rooms, and finally the second stage of the thermostat should activate the furnace. If the sun is at an intensity sufficient for collection, room heating from storage can be forced by disconnecting one of the leads from sensor S1.

HEAT EXCHANGER FOR SERVICE HOT WATER

Only one heat exchanger, a cross-flow air-to-water type such as described in Module 7, is required in an air-heating solar system in which service hot water is provided. Water circulation rates of 2 to 3 gpm are usually satisfactory for most domestic systems. Temperature rise in the heat exchanger will depend upon air temperature from the collectors, but a maximum rise of 15°F at midday with air temperature near 140°F will be satisfactory. Air flow rate through the heat exchanger is established by the area of collectors. With this information, manufacturers or their representatives should be able to recommend a heat exchanger size.

The heat exchanger should be in a location where it cannot freeze. By placing it in the heated space inside the building, there is reasonable protection from freezing. But if the exchanger is in the hot air duct between collector and blower (air handler), and if the collector damper in the air handler does not close tightly during the night heating mode from storage, cold air from the collector could flow through the duct and freeze the water in the heat exchanger. A better location for the heat exchanger is between the air handler and storage as shown

in Figure 8-5. Even if the collector damper is not tightly closed, cold air cannot come in contact with the heat exchanger in this location.

BLOWERS, AIR HANDLERS

PERFORMANCE CURVES

Representative performance curves for a centrifugal blower are illustrated in Figure 8-7. The pressure-discharge curve has a dip at 20 to 30 percent of peak capacity and a maximum at mid range. The brake horsepower (BHP) curve increases monotonically. Peak efficiency occurs to the right of peak pressure. For a blower that is belt-coupled to the motor, the speed of the impeller is variable. The effect of impeller speed on performance is illustrated by dashed curves, the rotational speed N_2 being less than N_1 .

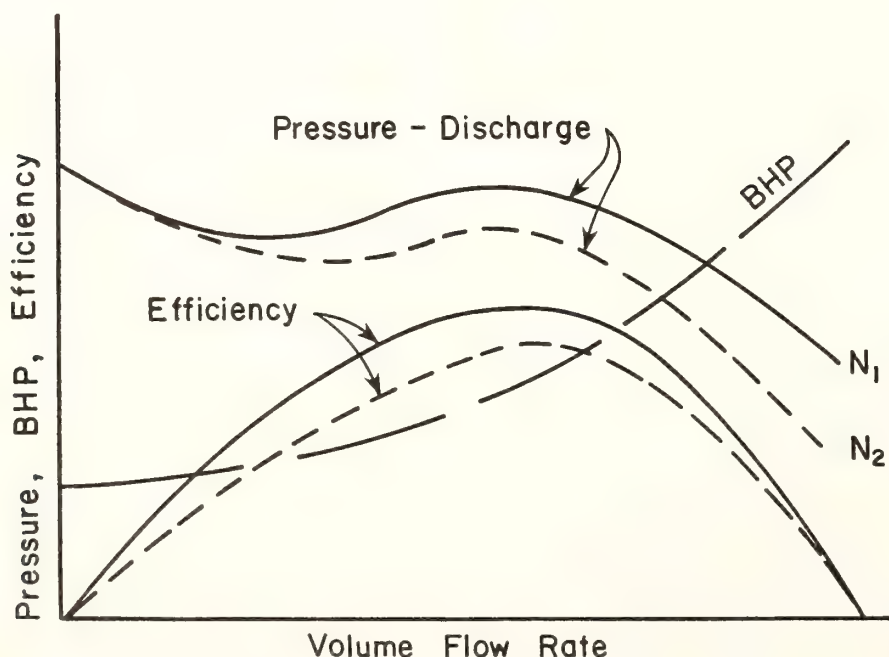


Figure 8-7. Typical Efficiency Curves for Blowers

BLOWER SELECTION

To select a blower, the pressure drop in the system and air flow rate are needed. The air flow rate is established by collector area, and the pressure drop in the system must be estimated. A rough layout of the ducts must be made and approximate sizes selected so that air flow velocity does not exceed about 600 feet per minute in the main circulation ducts.

Pressure drop through the collector for various flow rates will be provided by the manufacturer, but duct losses must be estimated. A graph for friction losses in ducts, shown in Figure 8-8, will assist in estimating the losses. Each mitered elbow or tee with turning vanes will cause an additional 0.005 in. water gauge pressure drop with air velocities of 600 ft per minute. Pressure drop through storage must be included.

With a single blower, the circulation rate through the distribution system must be adequate for heating the rooms, and the blower size selected for collecting solar heat must match room heating requirements. When the two circulation requirements are significantly different, a two-speed blower may be selected, one speed for collecting heat and another for distribution. As an approximate rule, if a system is sized to provide about 50 percent of the annual heating load, the air flow rate for the solar system will match the requirements for air circulation through the rooms.

Many residential and commercial air-heating solar systems have been provided with a blower and motorized dampers preassembled in a cabinet to which ducts are simply connected by the installer. These air handlers are commercially available in a one-blower configuration for

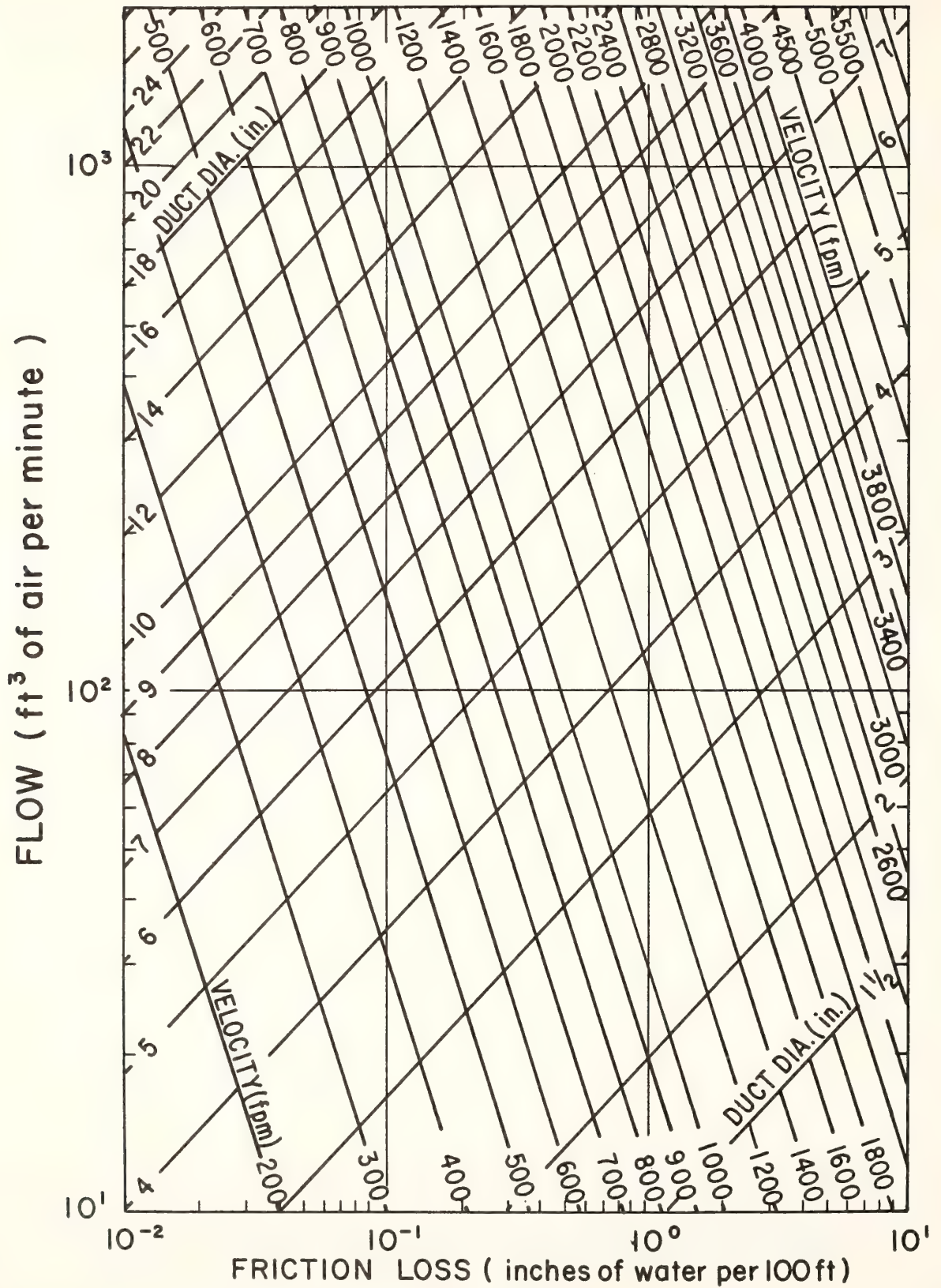


Figure 8-8. Friction Loss in a Straight Duct

systems such as described above, or in a design suitable for use with a second blower, usually part of a commercial warm air furnace. The latter system is more fully described in Module 10.

REFERENCES

1. Dickinson, W.S., Neifert, R.D., Löf, G.O.G., and Winn, C.B., (1975). "Performance Handbook for Solar Heating Systems". Presented at the 1975 International Solar Energy Society Congress, University of California at Los Angeles, California.
2. Klein, S.A., Beckman, W.A., and Duffie, J.A., (1975). "A Design Procedure for Solar Heating Systems". Presented at the 1975 International Solar Energy Society Congress, University of California at Los Angeles, California.
3. Duffie, J.A. and Beckman, W.A., (1974). Solar Energy Thermal Processes, John Wiley and Sons, New York.
4. Close, D.J., (1965). "Rock Pile Thermal Storage for Comfort Air Conditioning", Mechanical and Chemical Engineering Transactions of the Institution of Engineers, Australia, Vol. MC1, No. 1, May 1965.
5. Peltzman, E.C., "Differential Thermostats for Solar Energy Systems", Rho Sigma, Inc., 15150 Raymer Street, Van Nuys, CA 91405.
6. Kreith, F., (1966). Principles of Heat Transfer, International Textbook Co., Scranton, PA.
7. ASHRAE (1972) Handbook of Fundamentals, ASHRAE, New York.
8. ASHRAE (1972) Guide and Data Book, ASHRAE, New York.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 9

DOMESTIC HOT WATER SYSTEMS

SOLAR ENERGY APPLICATIONS LABORATORY
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OBJECTIVES

From the contents of this module the trainee should be able to:

1. Identify the types of domestic hot water systems available.
2. Select a type of solar domestic hot water system that is appropriate for a particular location.
3. Select a suitable collector area and storage tank size for a specific application.
4. Design a solar domestic hot water system and specify the installation requirements.

INTRODUCTION

The oldest and simplest domestic use of solar energy is for heating water. Solar hot water heaters were used in the United States at least 75 years ago, first in southern California and later in southern Florida. Although the use of solar water heaters in the United States declined during the last 40 years, use in Australia, Israel, and Japan has risen rapidly, particularly in the last 15 years. Since 1974, solar water heating is again increasing in the United States as a direct result of the general public interest in solar energy applications, the demonstration programs of public utility companies, and the Solar Demonstration Program and the research and development activities supported by the U. S. Department of Energy.

TYPES AND CHARACTERISTICS OF SOLAR HOT WATER HEATERS

Most of the solar hot water heaters that have been experimentally and commercially used can be placed in two main groups:

1. Non-circulating types, involving the use of water containers that serve both as solar collectors and storage.
2. Circulating types, involving the supply of solar heat to a fluid circulating through a collector and storage of hot water in a separate tank.

The circulating group may be divided into the following types and sub-types:

1. Direct heating, single-fluid types in which domestic water is heated directly in the collector, by:
 - (a) Thermosiphon circulation between collector and storage.
 - (b) Pumped circulation between collector and storage.
2. Indirect heating, dual-fluid types in which a non-freezing medium is circulated through the collector for subsequent heat exchange with water, when:
 - (a) Heat transfer medium is a non-freezing liquid.
 - (b) Heat transfer medium is air.

NON-CIRCULATING TYPE

Although of little potential interest in the United States, a type of solar water heater extensively used in Japan involves heat collection and water storage in the same unit. The most common type comprises a set of four to eight black plastic tubes about six inches in diameter and several feet long placed side-by-side in a box covered with glass

or clear plastic. Usually mounted in a tilted position, the tubes are filled each morning with water and heated by solar energy throughout the day. The filling can be accomplished by use of a float-controlled valve and a small supply tank. Late in the day, heated water can be drained from the tubes for bathing and other household uses not requiring pressurized hot water service.

Another type that can be used in non-freezing climates consists of a storage tank located inside a glazed and insulated box or cabinet placed at ground level outside the building. An insulated door on the south side of the box can be closed at night to reduce heat losses from the tank. The system operates with mains pressure so no pump is required. Tanks can be supported at a tilted angle to maximize exposure of the blackened sides to sunshine, or the tank may be placed horizontally and an insulated lid added to expose more of the tank wall to direct sunlight.

DIRECT HEATING, THERMOSIPHON CIRCULATING TYPE

The most common type of solar water heater in non-freezing climates is shown in Figure 9-1. The collector, usually single glazed, may vary in size from about 30 square feet to 80 square feet, and the insulated storage tank is commonly in the range of 40 to 80 gallons capacity. The hot water requirements of a family of four persons can usually be met by a system in the middle of this size range, in a sunny climate.

The thermosiphon system may be designed to operate at supply line pressure, or as an unpressurized system by installing a float valve in the storage tank. Alternatively a float-controlled elevated head tank can be utilized. For unpressurized systems, gravity flow from the hot water tank to hot water faucets would have to be accepted, although an automatic pump could be installed to provide pressure in the hot water

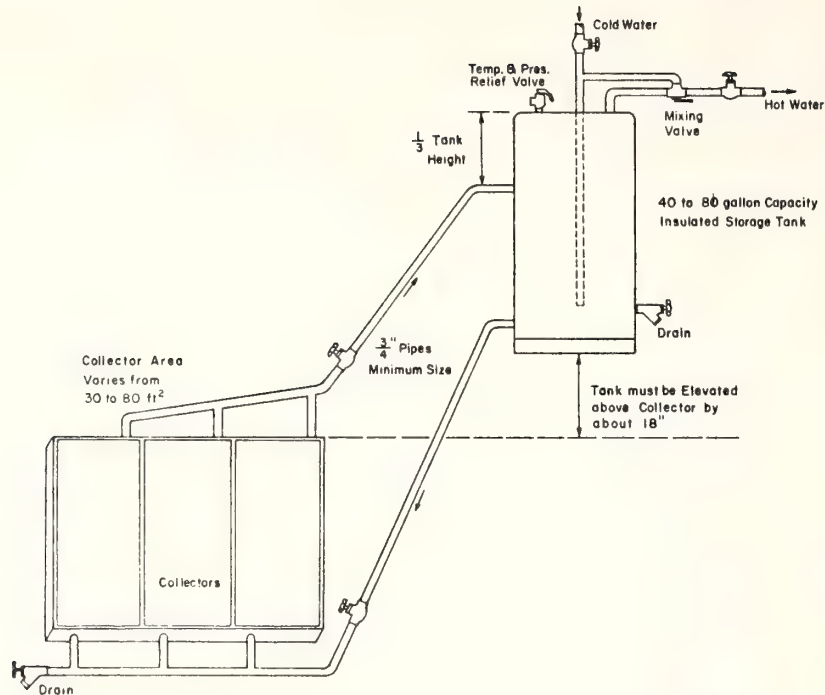


Figure 9-1. Direct Heating, Thermosiphon Circulating Solar Water Heater

supply line. Plumbing systems and fixtures in the United States normally require a pressurized system.

Locating the tank higher than the top of the collector permits circulation of water from the bottom of the tank through the collector and back to a point near the top of the tank. The density difference between cold and hot water produces the circulating flow. Temperature stratification in the storage tank permits operation of the collector under most favorable conditions, water at the lowest available temperature being supplied to the collector and the highest available temperature being provided to service. Circulation occurs only when solar energy is being received, so the system is self-controlling. The higher the radiation level, the greater the heating and the more rapid the circulating rate will be. In a typical collector under full sun, a

temperature rise of 15°F to 20°F is commonly realized in a single pass through the collector.

To prevent reverse circulation and cooling of stored water when no solar energy is being received, the bottom of the tank should be located above the top header of the collector. If the collector is on a sloping house roof, the tank may also be on the roof or in the attic space above the collector level.

Although seldom used in cold climates, the thermosiphon type of solar water heater can be protected from freezing by draining the collector. To avoid draining the storage tank also, thermostatically actuated valves in the lines between collector and storage tank must close when freezing threatens; a collector drain valve must open, and a collector vent valve must also be open. The collector will then drain and air will enter the collector tubes. Water in the storage tank, either inside the heated space or sufficiently well insulated to avoid freezing, does not enter the collector during the period when sub-freezing temperatures prevail. Resumption of operation requires automatic closure of the drain and vent valves and opening of the valves in the circulating line. The possibility of control failure or valve malfunction makes this complex system unattractive in freezing climates.

DIRECT HEATING, PUMP CIRCULATION TYPES

If placement of the storage tank above the collector is inconvenient or impossible, the tank may be located below the collector and a small pump used for circulating water between the collector and storage tank. This arrangement is usually more practical than the thermosiphon type in most residential buildings, because the collector would

often be located on the roof with a storage tank in the basement. Instead of thermosiphon circulation when the sun shines, a temperature sensor actuates a small pump which circulates water through the collector-storage loop. A schematic arrangement is shown in Figure 9-2.

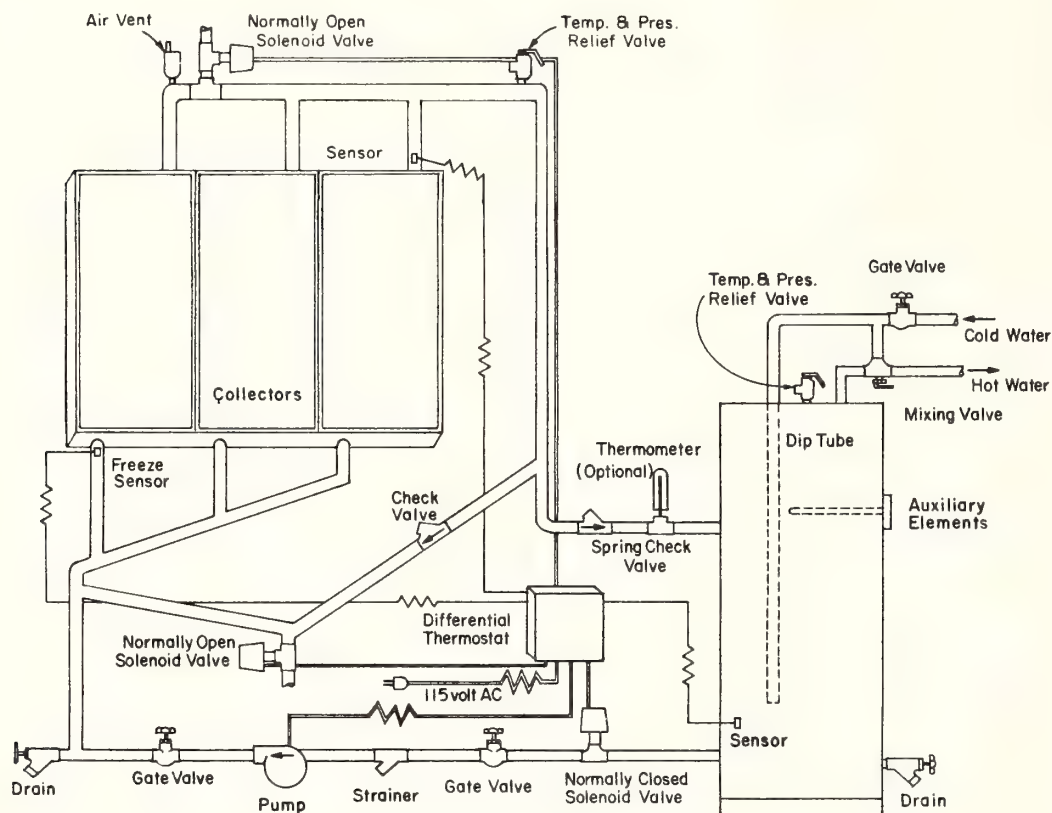


Figure 9-2. Direct Heating, Pump Circulation Solar Water Heater with Automatic Drain-Down (Applicable also to a Two-Tank System)

To obtain maximum utilization of solar energy, control is based on the difference in water temperature at collector outlet and bottom of storage tank. Whenever this difference exceeds a preset number of degrees, say 10°F , the pump motor is actuated. The sensor at the collector outlet must be located close enough to the collector so that it is affected by collector temperature even when the pump is not running. Similarly, the sensor in the storage tank should be located in or near

the bottom outlet from which the collector is supplied. When the temperature difference is less than a somewhat lower preset value, say 2°F to 5°F, the pump is shut off and circulation ceases. To prevent reverse thermosiphon circulation and consequent water cooling when no solar energy is being received, a check valve should be located in the circulation line.

If hot water use is not sufficient to maintain storage tank temperature at normal levels (as during several days of non-use), boiling may occur in the collector. If a check valve or pressure-reducing valve prohibits back flow from the storage tank into the main, a relief valve must be provided in the collector-storage loop. The relief valve will permit the escape of steam and prevent damage to the system.

DIRECT HEATING, PUMP CIRCULATION, DRAINABLE TYPES

If the solar water heater described above is used in a cold climate, it may be protected from freeze damage by draining the collector when sub-freezing temperatures are encountered. Several methods can be used, all of which must provide dependable drainage, even when electric power may not be available. One arrangement is shown in Figure 9-2.

Drainage of the collector in freezing weather can be accomplished by automatic valves which provide water outflow to a drain (sewer) and air inflow to the collector. Although not the most common control system, one arrangement operates so that whenever the circulating pump is not running, these two valves are open. To assure maximum reliability, the valves should be mechanically driven to the drain position (by springs or other means), rather than electrically, so that in the event of a power failure, the collector can automatically drain. A

disadvantage of this design is the daily exposure of collector tubing to air and its corrosive action.

The drainage system shown in Figure 9-2 is actuated by the temperature sensor ("Freeze Sensor") at the bottom of the collector. When it indicates a possibility of freezing, it causes the drainage and vent valves to open, thereby providing protection. Simultaneously with the opening of these valves, a motorized valve between the storage tank and pump inlet closes and a check valve in the line between collector and storage tank is forced shut, so that water pressure is retained in the tank. The temperature sensor can be of the vapor pressure type, with capillary tube connections to mechanical valve actuators, or of the electrical type where the valves are held open by electrical means, automatically closing at low temperatures and/or when electrical failure occurs.

Start-up of a vented collector system must permit the displacement of air from the collector. In either the line-pressure system or the unpressurized system, the entry of water into the collector (from the shut-off valve or pump) forces air from the collector to the atmosphere through a simple air bleed valve of a type which automatically passes air but shuts off when water reaches it. The vent valve which admits air to the collector may be electrically operated (such as a normally open solenoid type), or it may be a "vacuum breaker" valve which is forced open by the decrease in pressure resulting from opening the drain valve.

Reduction in the number of automatic valves required in this system, with at least two activators, is being attempted through design of one automatic valve unit, with multiple ports, stems, and seats, which can perform all the functions of the several valves now commonly

used. Considerable simplification, cost reduction, and reliability improvement may then be achieved.

CIRCULATING TYPE, INDIRECT HEATING

The needs and means for collector drainage of direct heating systems in freezing climates involve added costs, and there is still a risk of freezing. The drainage requirement can be eliminated by the use of a non-freezing heat transfer medium in the solar collector and a heat exchanger for transfer of heat from the solar-heated collecting medium to the service water. The collector never needs to be drained, and there is little risk of freezing. Corrosion rate in the wet collector tubes is also decreased because they are always liquid-filled and there is no free oxygen in the heat transfer medium.

Liquid Transfer Media

A method for solar water heating with a liquid heat transfer medium in the solar collector is illustrated in Figure 9-3. The most commonly used liquid is a solution of ethylene glycol (automobile radiator anti-freeze) in water. A pump circulates this unpressurized solution, as in the direct water heating system, and delivers the liquid to and through a liquid-to-liquid double-wall heat exchanger. Simultaneously, another pump circulates domestic water from the storage tank through the exchanger, back to storage. The control system is essentially the same as that in Figure 9-2. If the heat exchanger is located below the bottom of the storage tank, and if the pipe sizes and heat exchanger design are adequate, thermosiphon circulation of water through the heat exchanger can be used and the pump can be eliminated from the water loop. A small expansion tank needs to be provided in the collector loop, preferably near the high point of the system if in an unpressurized collector circuit. This tank must have a vent to the atmosphere. If the

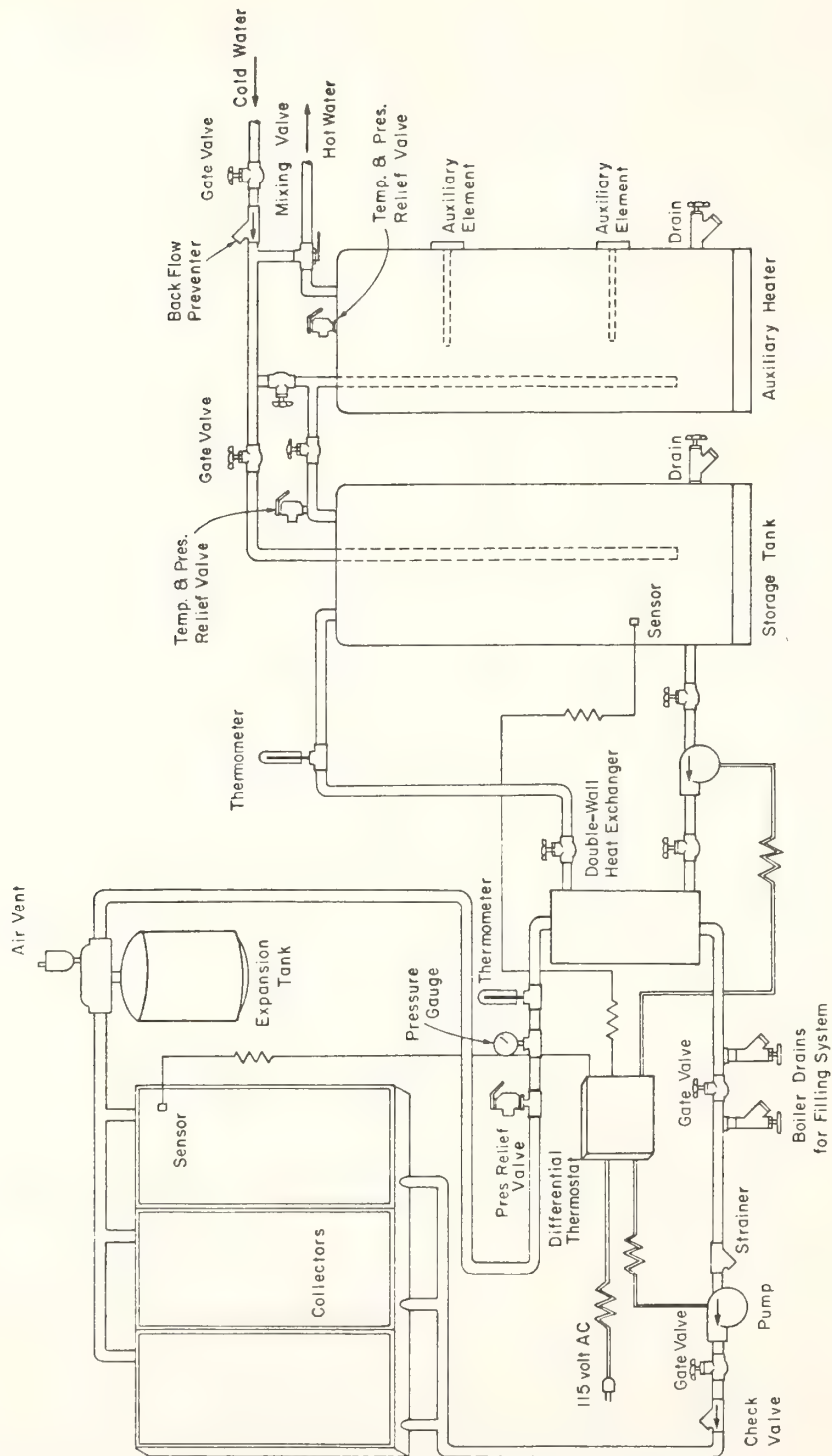


Figure 9-3. Indirect Heating, Pump Circulation Solar Water Heater with Liquid Heat Transfer Media

collector loop is operated under positive pressure, an expansion tank containing a diaphragm balanced by air pressure may be located at a convenient point, usually in the line near the solution pump.

To meet most code requirements, the heat exchanger must be of a design such that rupture or corrosion failure will not permit flow from the collector loop into the domestic water, even if pressure on the water side of the exchanger drops below that on the antifreeze side. A conventional tube-and-shell exchanger would therefore not usually be acceptable. Similarly, a single pipe coil inside the storage tank, through which the collector fluid is circulated, would not be satisfactory. Parallel tubes with metal bonds between them, so that perforation of one tube could not result in liquid entry into the other tube, would be satisfactory. A finned tube air-to-liquid heat exchanger can also be used by circulating the two liquids through alternate rows of tubes, heat transfer being by conduction through the fins. Tubes within tubes, with slight clearance between them for outflow of leaked fluid, may be used as immersed coils.

Although aqueous solutions of ethylene glycol and propylene glycol appear to be most practical for solar energy collection, organic liquids such as silicone oil, Dowtherm J[®] and Therminol 55[®] may be employed. Price and viscosity are drawbacks, but chemical stability and assurance against boiling are advantages over the antifreeze mixtures.

Air Transfer Media

An air-heating collector can be used to heat domestic water with an air-to-water heat exchanger, as illustrated in Figure 9-4. A solar air heater is supplied with air from a blower, the air is heated by passage

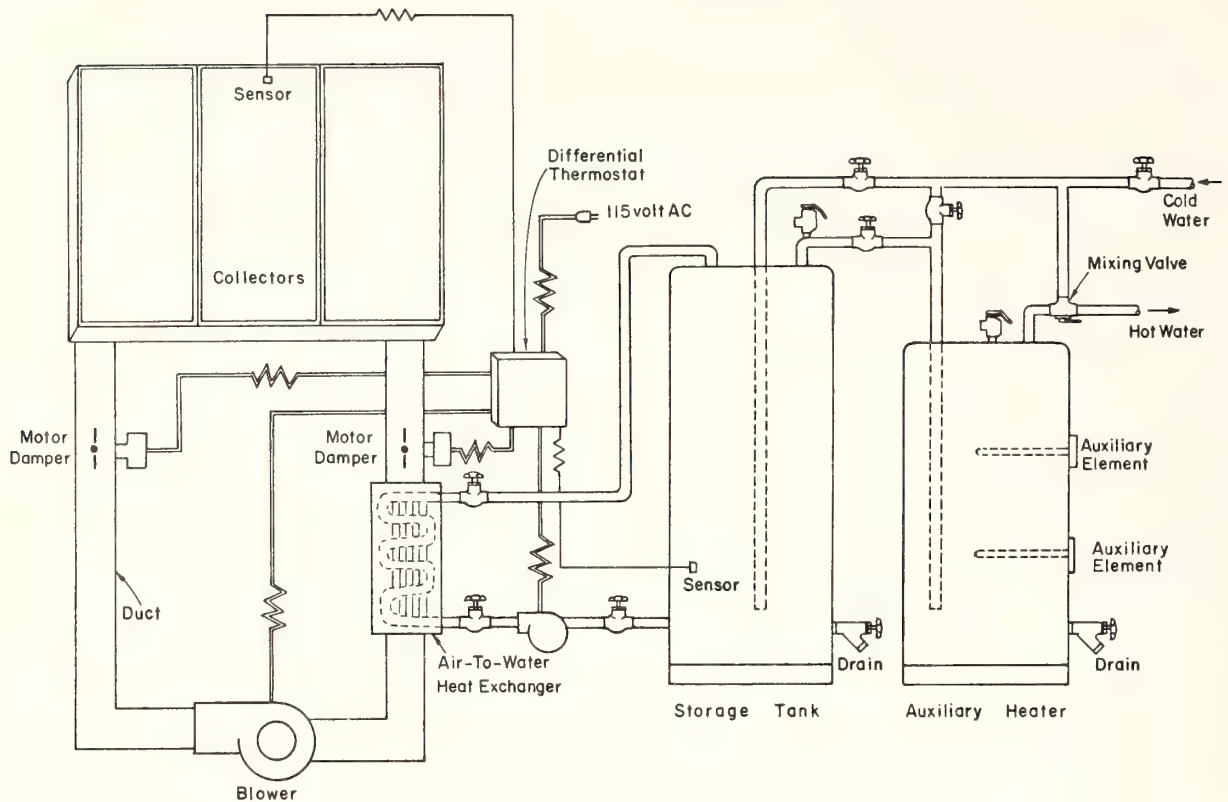


Figure 9-4. Solar Water Heater with Air Collectors

through the collector, and the hot air is then cooled in the heat exchanger through which domestic water from a storage tank is either pumped or circulated by thermosiphon action. Air from the heat exchanger is recirculated to the collector. Differential temperature control (between collector and storage) is employed as in the other systems described. Advantages of the air heat transfer medium are the absence of corrosion in the collector loop, freedom from liquid leakage, and freedom from freezing, boiling, and evaporation of collector fluid. Disadvantages are the larger conduit between collector and heat exchanger, higher power consumption for circulation, and larger collector surface requirements compared to liquid-heating collectors.

AUXILIARY HEAT

A dependable supply of hot water requires the availability of auxiliary heat for supplementing the solar source. The numerous methods of providing auxiliary heat vary in cost and effectiveness. A general principle for maximizing solar supply and minimizing auxiliary use is to avoid direct or indirect auxiliary heat input to the fluid entering the solar collector. If auxiliary heat input is added to the solar hot water storage tank, so that the temperature of the liquid supplied to the collector is increased above that which only the solar system would provide, efficiency is reduced because of higher heat losses from the collector. Thus, auxiliary heat should be added at a point beyond (downstream from) the solar collector-storage system. A conventional electric hot water heater is shown in Figures 9-3 and 9-4 supplied with hot water from the solar tank (whenever a hot water tap is opened). Any deficiency in temperature is made up by electricity in the thermostatted conventional heater. To avoid confusion in terminology, the first, or solar heated, tank is commonly referred to as the "pre-heat" tank or "solar pre-heat" tank, and the second one as the "auxiliary hot water heater". It is evident that auxiliary heat supply in these designs cannot adversely affect the operation of the solar system.

A one-tank system with the electric resistance heaters in the upper portion of the solar storage tank, as shown in Figure 9-2, is cheaper than the two-tank type, and should have nearly the same solar collection efficiency. Temperature stratification in the tank, accomplished by bringing cold water from the main into the bottom and by circulating through the collector from the bottom of the tank to the upper portion

of the tank, prevents auxiliary heat from increasing the temperature of the water supplied to the collector. Water returning from the collector may be brought into the tank below the level of the resistance heater so that a supply of hot water is always available at the thermostatted temperature. In effect, the two tanks shown in Figures 9-3 and 9-4 are combined into one, with temperature stratification providing a separation. The total amount of storage is reduced unless the one tank is increased in size, but heat losses are also reduced. If relatively high-temperature water is desired, there may be an undesirable influence of auxiliary supply on collector efficiency because of some mixing in the tank.

If auxiliary heating is by gas or oil, a two-tank system must be used because burners are located at the bottom of the heater tank. Auxiliary heat can thus raise the temperature of water near the bottom of the tank and adversely affect collector efficiency if the fuel-fired water heater is also used as the solar supply tank.

Although the description of the above system refers to direct circulation of water through the collector, the same factors apply to the systems involving heat exchange with antifreeze solutions or air circulating through the collector. In all cases, auxiliary heat should be supplied downstream from the solar heat supply, regardless of whether the water itself is circulated through the collector or whether heat is exchanged between the domestic water and a solar heat transfer fluid.

TEMPERATURE STRATIFICATION IN SOLAR HOT WATER TANK

As in a conventional hot water heater, the temperature in the upper part of a solar hot water tank will normally be considerably higher

than at the bottom. The lower density of hot water permits this stratification, provided that turbulence at inlet and outlet connections is not excessive. The supply of relatively cold water from the bottom of the tank to the collector permits the collector to operate at its highest possible efficiency under the prevailing ambient conditions. With a circulation rate such that a temperature rise through the collector of 15°F to 20°F occurs, water from the lower part of the storage tank is furnished to the collector for maximum effectiveness. If little hot water is withdrawn from the tank during a sunny day, the late afternoon temperature at the bottom of an 80-gallon tank connected to a 40- to 50-square-foot collector may be well above 100°F -- even approaching the temperature at the top of the tank. Collection efficiency thus varies throughout the day, depending not only on solar availability but also on the temperature of water supplied to the collector from the tank bottom. Data presented at the end of this module illustrate the range of temperatures achievable in solar water heaters.

TEMPERATURE CONTROL LIMIT

In addition to the differential temperature control desirable in most solar water heating systems (which sense temperature differences between collectors and storage), protection against excessive water temperature may be necessary. Several possible methods can be used. In nearly all types of systems, whether direct heating of the potable water or indirect heating through a heat exchanger, a thermostatically controlled mixing valve (tempering valve) can be used to provide hot water at constant temperature for household use as shown in Figures 9-3 and

9-4. Cold water is admitted to the hot water line immediately downstream from the auxiliary heater in sufficient proportion to secure the desired preset temperature. The solar hot water tank is allowed to reach any temperature attainable, and the auxiliary heater furnishes additional energy only when the auxiliary tank temperature drops below the thermostat set point. Maximum solar heat delivery is thus achieved, and no solar heat needs to be discarded except that which might sometimes be delivered when the solar preheat tank is at the boiling point. Any additional solar heat collected under that condition would be dumped through a relief valve with steam escaping to the surroundings.

Venting of steam from a solar hot water system involving a dual-liquid design, with heat exchanger, should normally be through a temperature-actuated valve in the hot water loop. Loss of collector fluid by boiling and vaporization is thereby avoided. It is necessary, however, in this design, that the boiling temperature of the collector fluid be at least 20°F higher than the temperature at which the steam vent valve in the storage loop is actuated. If, for example, the relief valve at the top of the storage tank opens at 210°F, so that hot water and steam are discharged, the glycol solution should not boil at 230°F. Maintenance of a 50-percent mixture will prevent boiling as long as the liquid is circulated, even if not pressurized. A pressure of 15 psi on the glycol solution, commonly provided, further protects it from boiling and loss through the safety vent on the collector loop. If an organic liquid such as silicone oil or Dowtherm J[®] is used as the collector fluid, there is no risk of boiling in the collector even in an unpressurized collection circuit. The temperature in the storage tank must be limited as in the case above.

Another option for high-temperature protection is available if an organic liquid or air is used as the heat collection medium. To prevent the storage tank from reaching a temperature higher than desired, a limiting thermostat in that tank can be used simply to discontinue circulation of the heat transfer fluid (organic liquid or air) through the collector and heat exchanger. No additional heat is therefore supplied to the hot water, and the intercepted solar energy is dissipated in the form of collector heat loss. The collector temperature rises substantially, frequently above 300°F, but with proper design, there should be no damage. With a reliable limit switch in the storage tank, there can be no dangerous pressure developing anywhere in the system. In addition, there is no loss of water (in the form of steam) even when there is no use of hot water for long periods.

If a hot water/cold water mixing valve downstream from the auxiliary heater is not provided, a temperature limit control in the solar storage tank can be set at the maximum desired temperature of service hot water. Water cannot then be delivered at a temperature higher than the set point in the solar storage tank or the set point in the auxiliary heater, whichever is higher. Less solar storage capability would be involved in this design because the solar storage tank is prevented from achieving higher temperatures, even when solar energy is available.

In a drain-down type of solar water heater operating at line pressure, with potable water circulating through the collector, the air bleed valve at the top of the collector may be able to vent sufficient steam to avoid overheating service water when usage is small. A temperature-actuated relief valve, set at a temperature or 210°F or less,

should also be provided in the top of the solar preheat tank to allow additional discharge of steam and hot water, if necessary.

LOCATION OF COLLECTORS

If the slope and orientation of a roof are suitable, the most economical location for a solar collector in a residential water heating system is on the south-facing portion of the roof. The cost of a structure to support the collector is thereby eliminated, and pipe or duct connections to the conventional hot water system are usually convenient. In new dwellings, most installations can be expected on the house roof. Even in retrofitting existing dwellings with solar water heaters, a suitable roof location can usually be provided.

If the mounting of collectors on the roof is impractical, for any of several reasons, a separate structure adjacent to the house may be used. A sloping platform supported on a suitable foundation can be the base for the collector. Pumps, storage tank, and heat exchanger, if used, can be located inside the dwelling. Effective insulation on ducts and piping must be provided, however, so that cold-weather operation will not be handicapped by excessive heat losses. In cold climates, collectors in which water is directly heated must be located so that drainage of the collector and exterior piping can be dependably and effectively accomplished.

ORIENTATION AND TILT OF COLLECTORS

When roof orientation and slope are not ideally suited for collector mounting, i.e., roof does not face due south and is not tilted

at the latitude angle, roof mounting may still be satisfactory. While collectors should be oriented to face due south whenever possible, variations as much as 15 degrees east or west of due south will have only slight effect on system performance. If the collectors are subject to shading in the late afternoon (say after 2:30 pm), orienting the collectors 15 degrees to the east will be beneficial to total heat collection during the day. Similarly, if morning cloudiness usually prevails because of local climatic conditions, it may be beneficial to face the collectors a few degrees to the west of south.

For maximum annual heat collection, a south-facing collector should be tilted at about latitude angle. However, variations of 10 degrees greater or less than latitude angle will generally not decrease the amount of heat collected by more than 5 percent.

PERFORMANCE OF TYPICAL SYSTEMS

GENERAL REQUIREMENTS

A typical family of four persons, in the United States, requires about 80 gallons of hot water per day. At a customary supply temperature of about 140°F and a cold water inlet temperature of 60°F, this is about 55,000 Btu per day.

There is a wide variation in the solar availability from region to region and from season to season in a particular location. There are also the short-term radiation fluctuations due to cloudiness and the day-night cycle.

Seasonal variations in solar availability result in a 200 to 400 percent difference in the solar heat supply to a hot water system. In

the winter, for example, an average recovery of 40 percent of 1200 Btu/ft² of solar energy on a sloping surface would require approximately 100 ft² of collector for an average daily requirement of 50,000 Btu. Such a design would provide essentially all of the hot water needs on an average winter day, but would fall short on days of less than average sunshine. By contrast, a 50-percent recovery of an average summer radiant supply of 2000 Btu/ft² would involve the need for only 50 ft² of collector for satisfying the average hot water requirements.

It is evident that if 50 ft² of collectors were installed, it could supply the major part, perhaps nearly all, of the summer hot water requirements, but it could supply less than half the winter needs. If 100 ft² of collectors were used to meet more of the winter demand, the system would be oversized for summer operation and excess solar heat would have to be wasted. In such circumstances, if an aqueous collection medium were used, boiling of the system would frequently occur and collector or storage venting of steam would have to be provided.

The more important disadvantage of the oversized collector (for summer operation) is the economic penalty associated with investment which is not fully utilized. Although the cost of a 100 ft² collector system would be approximately double that of a 50 ft² unit, the annual useful heat delivered would be considerably less than double. The larger system would, of course, deliver about twice as much heat in the winter season, when nearly all of it could be used, but in the other seasons, particularly in summer, heat overflow would occur. The net effect of these factors is a lower economic return per unit of investment for the larger system. Stated another way, more Btu per dollar of investment (hence cheaper solar heat) can be delivered by the smaller system.

As a conclusion to the above example, practical design of solar water heaters should be based on desired hot water output in the sunniest months rather than at some other time of year. If based on average daily radiation in the sunniest months, the unit will be slightly oversized and a small amount of heat will be wasted on days of maximum solar input. And quite naturally, on partly cloudy days during the season, some auxiliary heat must be provided. In the month of lowest average solar energy delivery, typically one-half to one-third as much solar-heated water can be supplied, or equivalently the same quantity of water is supplied but with a temperature increase above inlet only one-half to one-third as high. Thus, fuel requirements for increasing the temperature of solar-heated water to the desired (thermostatted) level could involve one-half to two-thirds of the total energy needed for hot water heating in a mid-winter month.

QUANTITATIVE PERFORMANCE

Although hundreds of thousands of solar water heaters have been used in the United States and abroad, quantitative performance data on systems in practical use have seldom been obtained. In households where no auxiliary heat was used, the solar system probably supplied hot water most of the time, but failed during bad weather. If booster heat was used, hot water was always available, but the relative contributions of solar and auxiliary were not known.

In a few research laboratories, particularly in Australia, some analytical studies of solar water heater performance, confirmed in part by experimental measurements, have been performed. More recently, analytical studies at the University of Wisconsin have been carried out,

and performance data on typical heaters have been obtained at the National Bureau of Standards. Table 9-1, based on an Australian study, shows the performance of a double-glazed, 45 ft² solar water heater at several locations in that country where climatic conditions are similar to those in parts of the U.S. Variable solar energy and ambient temperature throughout the year result in 1.4 to 2.5 times as much solar heat supply to water in summer as in winter. Climatic differences produced a solar heat percentage ranging from 60 percent to 81 percent of the annual total hot water requirements. Table 9-2 shows monthly performance of the same system in Melbourne, Australia, with average collection efficiency varying between 29 and 40 percent of incident radiation. Variation in inlet, outlet, and ambient temperature in a typical thermosiphon type of solar water heater is shown in Figure 9-5.

In a simulation study at the University of Wisconsin, hot water usage was programmed for a hypothetical residential user. The results show only slight variation in solar heat utilization at several use schedules and indicate only minor influence of storage temperature stratification on collector efficiency.

In summary, the normal output of well-designed solar water heating systems can be roughly estimated by assuming approximately 40 percent solar collection efficiency. Average monthly solar radiation multiplied by collector area and 40 percent delivery efficiency can provide a rough measure of daily or monthly Btu delivery. The total Btu requirements for the hot water supply, based on the volume used and the temperature increase set, then serves as the basis for computing the percentage contribution from solar and the portion required to be supplied by fuel or electricity.

Table 9-1

Daily Means for Twelve Consecutive Months of Operation of Solar Water Heaters at Various Localities

Location	Adelaide	Brisbane*	Canberra	Denili- quin	Geelong	Melbourne	Sydney
Hot water discharge**(gallons, US)	54.2	54.6	51.4	50.9	50.4	54.6	53.9
Electrical energy consumed (kWh)	3.5	2.5	3.4	2.5	3.8	4.6	4.4
Cold water temperature (°C)	17.7	21.6	12.7	16.8	15.9	16.1	16.6
Hot water temperature (°C)	58.9	56.4	58.4	60.3	58.7	57.4	57.7
Energy required to heat water (kWh)	9.8	8.4	10.3	9.7	9.5	9.9	9.8
Heat loss from storage tank (kWh)	2.2	1.9	2.5	2.5	2.2	1.9	1.9
Total energy consumed (kWh)	12.0	10.3	12.8	12.2	11.7	11.8	11.7
Solar energy contributed (kWh)	8.5	7.8	9.4	9.7	7.9	7.2	7.3
Solar energy contributed (percent)	71.0	76.0	73.0	81.0	67.0	61.0	62.0
Solar contribution best month (%)	99.0	94.0	98.0	100.0	92.0	95.0	70.0
Solar contribution worst month (%)	47.0	57.0	43.0	57.0	45.0	38.0	51.0
Ratio best to worst	2.1	1.6	2.3	1.8	2.0	2.5	1.4

* Hail screens suspended above the absorbers. No correction made for reduction of absorbing area.

** Water discharged at 6:00 a.m. daily. Double-glazed, flat-black, 45 ft² solar collector tilted toward equator at latitude angle plus 2.5 degrees. Storage 84 gallons (US). Thermosiphon circulation. Electric auxiliary heat.

Table 9-2

Solar Water Heater Performance in Melbourne, Australia

Month	Mean Insolation on Absorber	Mean Daily Supplemen- tary Energy	Mean Daily Solar Energy Contribution		System Efficiency
	Btu/ft ² ·day	kWh	Percent	kWh	Percent
January	1630	2.9	75	8.9	40
February	2220	0.5	95	9.5	32
March	1690	2.6	74	7.4	33
April	1240	5.2	52	5.6	34
May	1290	6.2	47	5.5	32
June	1220	7.7	39	4.9	30
July	1290	8.1	38	5.0	29
August	1530	6.1	50	6.1	30
September	1600	4.9	59	7.1	33
October	1860	3.9	67	7.9	32
November	1880	3.7	68	7.9	32
December	1790	3.5	72	9.0	38
Year	1610	4.6	61	7.2	35

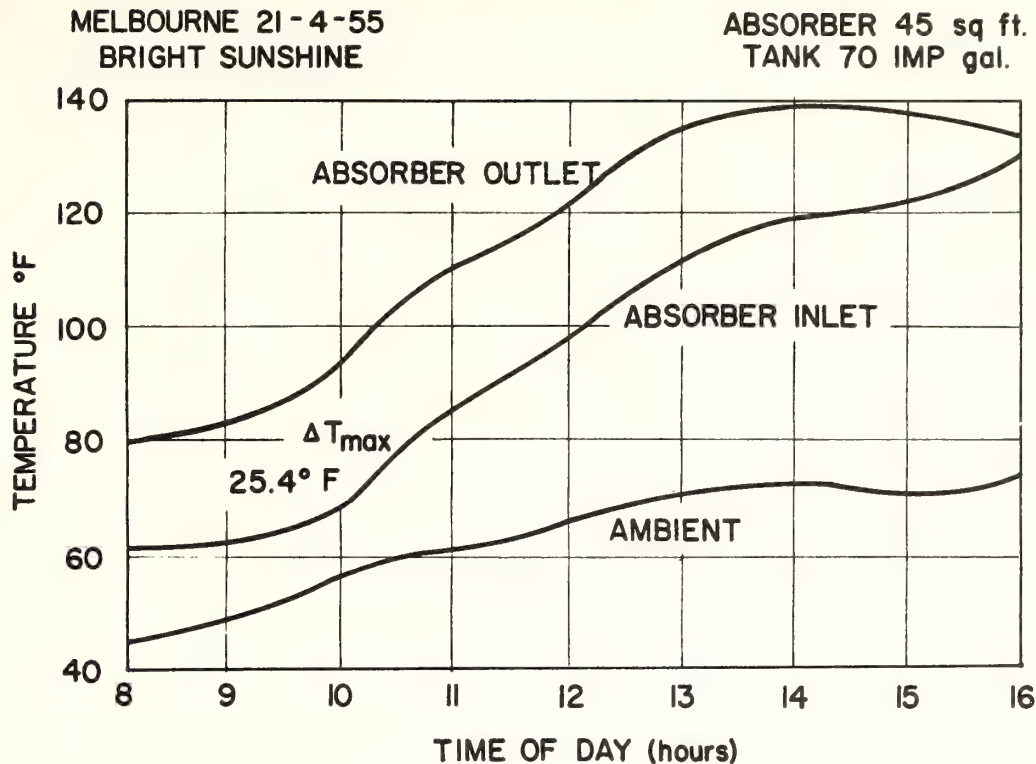


Figure 9-5. Absorber and Tank Temperatures for Thermosiphon Flow During a Typical Day

SIZING THE COLLECTORS

The curves shown in Figure 9-6 may be used to estimate the solar collector size required for hot water service in residential buildings having typical hot water systems. The system is assumed to be a pumped liquid type, with a liquid-to-liquid heat exchanger, delivering hot water to scheduled residential uses from 6:00 a.m. until midnight. A collector, of good quality (but not with the highest efficiency available), is mounted at a tilt angle equal to the latitude; water is heated by collector and auxiliary from 50°F to 140°F. The shaded band represents results of computer calculations for eleven different locations in the United States. The cities included in the study are Boulder, Colorado; Albuquerque, New Mexico; Madison, Wisconsin; Boston, Massachusetts; Oak Ridge, Tennessee; Albany, New York; Manhattan,

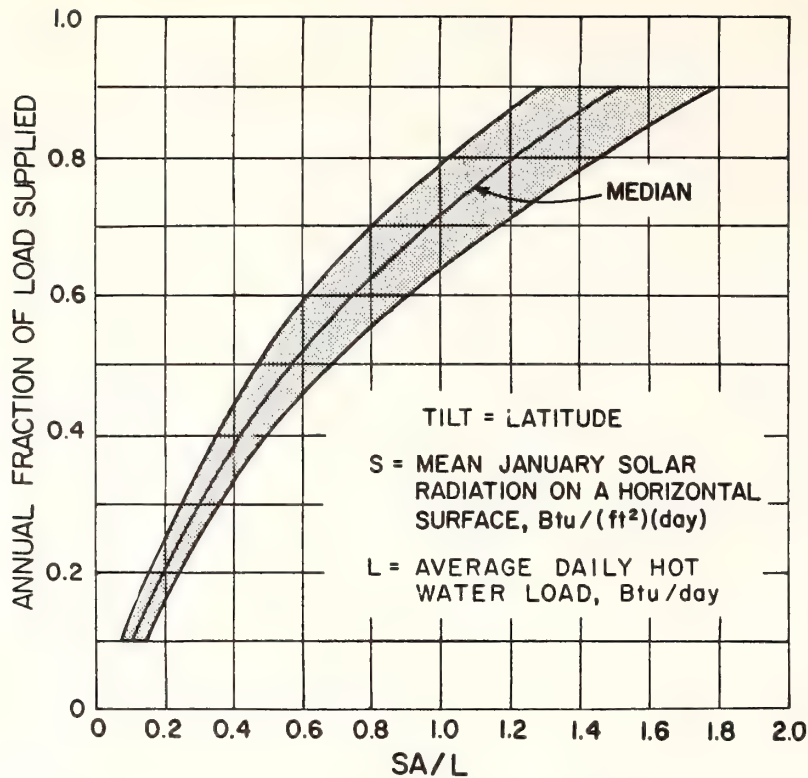


Figure 9-6. Fraction of Annual Load Supplied by Solar as a Function of January Conditions for Hot Water Heaters

Kansas; Gainesville, Florida; Santa Maria, California; St. Cloud, Minnesota; and Washington, D.C.

The hot water loads used in the computations range from 50 gallons per day (gpd) to 2000 gpd. The sizing curves are approximate and should not be expected to yield results closer than 10 percent of actual value.

The vertical axis shows the fraction of the annual water heating load supplied by solar. The horizontal axis shows values of the parameter, $S_j A/L$, which involves the average daily January radiation on a horizontal surface, S_j ; the required collector area, A , to supply a certain percentage of the daily hot water load, L . The January average daily total radiation at locations in the United States can be estimated from the radiation map in Figure 9-7. Values on the map are given in Btu/(ft²·day). The curves are not applicable for values of solar fraction, f , greater than 0.9.

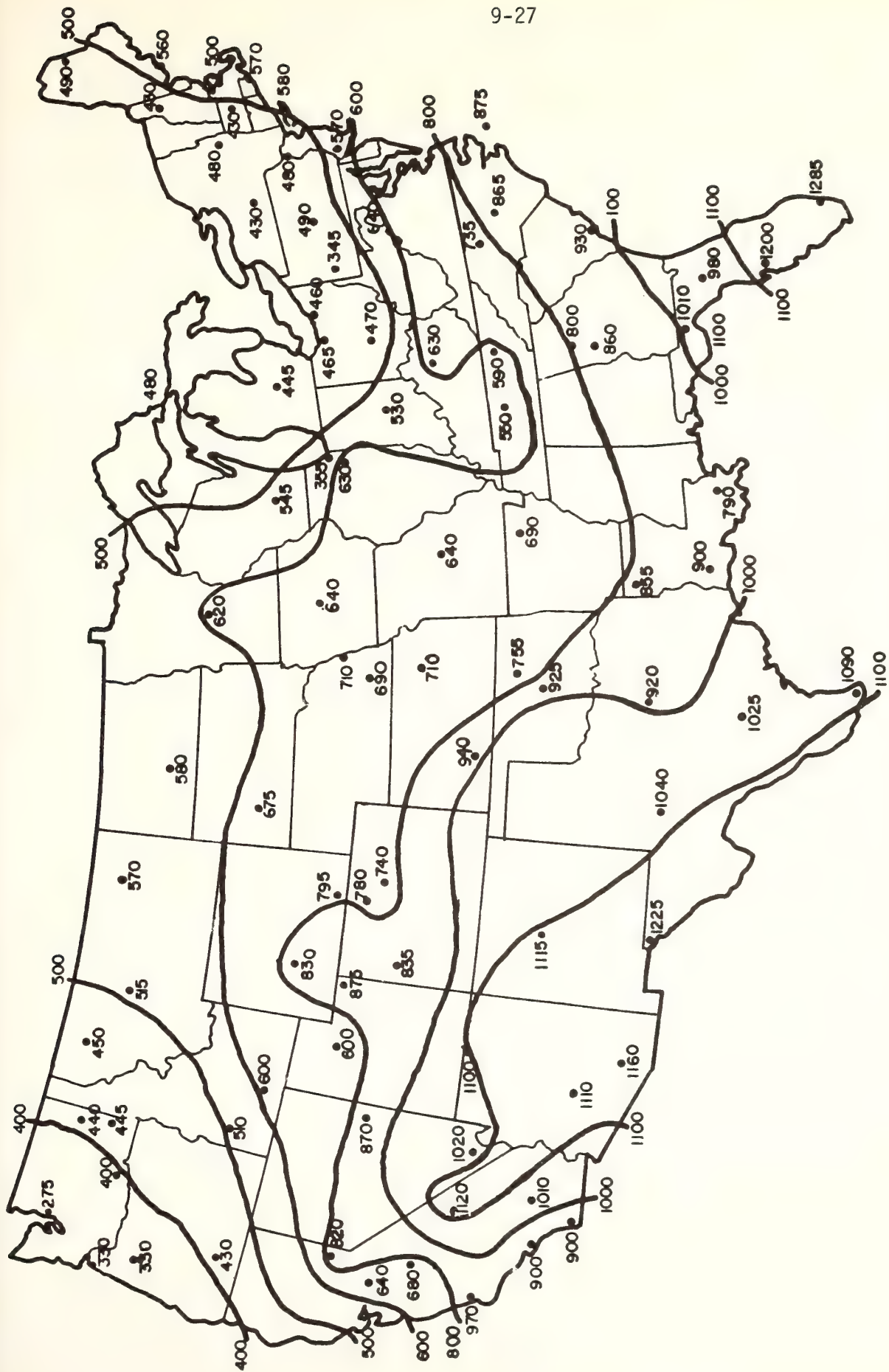


Figure 9-7. Average Daily Solar Radiation (Btu/ft²), Month of January

It should be remembered that the service hot water load will be nearly constant throughout the year while the solar energy collected will vary from season to season. A system sized for January, with collectors tilted at the latitude angle, will deliver high temperature water and may even cause boiling in the summer. A system sized to meet the load in July will not provide all of the load in the winter months. Tilting of the collector partially overcomes the effect of month-to-month differences in radiation and temperature.

SIZING EXAMPLES

Example 9-1

Determine the approximate size of collector needed to provide hot water for a family of four in a residential building in Kansas City, Missouri.

SOLUTION: The average daily service hot water load in January is:

$$L = 80 \text{ gallons/day} \times 8.34 \text{ pounds/gallon}$$

$$\times 1 \text{ Btu/(lb)}(^{\circ}\text{F}) \times (140^{\circ}\text{F} - 50^{\circ}\text{F}) = 60,048 \text{ Btu/day}$$

The desired service water temperature is 140°F and the temperature of the cold water from the main is 50°F . The monthly average daily solar radiation, S_J , available in January, from Figure 9-7, is about $680 \text{ Btu}/(\text{ft}^2 \cdot \text{day})$. For a water system to provide 60 percent of the annual load, from Figure 9-6, $S_J A/L$ is about 0.8. Therefore:

$$A = 0.8 \times L/S_J = (0.8 \times 60048)/680 = 70.6 \text{ ft}^2.$$

If 3-by-8-foot collector modules are available, 2.9 units would be required. Three collector units should therefore be used.

Example 9-2

Determine the size of collector needed to provide hot water for a family of four in Albuquerque, New Mexico.

SOLUTION: The monthly load will be approximately the same as in Example 9-1:

$$L=60,048 \text{ Btu/day}$$

From Figure 9-7, $S_j = 1115 \text{ Btu}/(\text{ft}^2 \cdot \text{day})$. For a system to provide 60 percent of the annual load, Figure 9-6 shows that $S_j A/L$ is approximately 0.8. The collector area required is:

$$A = (0.8 \times 60048)/1115 = 43.1 \text{ ft}^2$$

Using 3-by-6-foot collector modules, 2.4 units would be required for this system; either two or three modules should be used. If two modules are used, the system would be expected to provide less than 60 percent of the annual load.

REFERENCES

1. Hollander, P., M. O'Neil and W. Fisher (1978) Installation Guidelines for Solar DHW Systems In One and Two Family Dwellings. Franklin Research Center.
2. U. S. Department of Energy (1978) "Solar Heating and Cooling Project Experiences Handbook" (preliminary issue). Prepared jointly by U. S. Department of Energy Project Management Centers, ASHRAE and the University of Alabama, July.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 10

SOLAR SYSTEMS FOR SPACE HEATING

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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OBJECTIVE

The objective of this module is to detail the design, operation, and performance of the principal types of solar heating systems. The trainee should be able to:

1. Determine the heating requirements of buildings and select solar heating systems best suited to those requirements.
2. Develop schematic and working plans of solar heating systems.
3. Specify compatible system components.
4. Determine the proper size of system components.
5. Provide design guidance and advice to installers of solar heating systems.

INTRODUCTION

In Module 2, the general principles of solar heating are presented. Subsequent modules contain design and operating details on solar collectors and other system components; solar systems for service hot water are covered in Module 9. The purpose of this module is the integration of the preceding information on components and sub-systems.

Successful application of solar heating requires careful selection of components, their proper sizing, and their skillful assembly into well-functioning systems. Collectors, heat storage units, pumps and fans, controls, heat exchangers, and auxiliary heaters must be effectively integrated. A practical solar system must function automatically, provide the desired comfort level in the building at all times,

require little maintenance, and operate reliably over a long period of time.

After selecting a system type and specific components, the designer must determine the collector area requirements and storage volumes needed for providing the desired portion of the annual heat demands. Air and liquid flow rates, and blower and pump sizes can then be established. Heat exchangers are selected in accordance with the required heat transfer rates and the exchanger characteristics. The auxiliary furnace is sized by conventional methods to meet the design heating load of the building.

SYSTEM TYPES

General descriptions of solar heating systems employing flat-plate liquid and air collectors are presented in Module 2. Although there is a great deal of variety in system design among these two main types, most of the practical installations can be classified into similar groups and sub-groups.

Liquid collectors are used in drain-back systems, with and without siphon return of water to storage, and in dual-liquid designs with a heat exchanger for transfer of heat from a non-freezing collector liquid to water storage. If heat is distributed in a hydronic system, auxiliary heat is usually supplied by use of a water boiler, but sometimes a liquid-to-liquid or air-to-liquid heat pump is used. If heat is distributed in warm air, a water-to-air exchanger is used for the solar supply, and a warm-air furnace is usually employed as auxiliary. An air-to-air heat pump may be used instead of the auxiliary furnace.

Most practical air solar systems involve some type of air-heating solar collector, a vertical-flow pebble bed for heat storage, and an air-to-water heat exchanger for service hot water supply. Variations in system design are usually limited to the air moving equipment (blowers and dampers) and auxiliary heat supply. A single blower and four motorized dampers are used in an air handler in numerous residential air solar systems, whereas two blowers and two motorized dampers are used in many others. A warm air furnace or air-to-air heat pump nearly always provides auxiliary heat.

LIQUID SYSTEMS

There are two main types of liquid solar heating systems in common use: one in which water is the heat collecting fluid, and the other which involves use of two liquids, one for collection and the second for storage. They differ in the principle applied to protect the collectors from freezing during cold sunless periods. There are also several arrangements for distributing heat from storage to the living space. The use of forced warm air from central heat exchangers, and hydronic loops with various types of exchangers in each room, are the most common methods. Additional variations involve the type of auxiliary heater, its energy supply, and its position in the heating system. The way in which service hot water supply is integrated with the solar heating system, and the methods for system control in winter and summer are further design and operating options.

SINGLE-LIQUID (DRAIN-BACK) SYSTEMS

Two designs for solar heat collection and storage involving water as the heat collection and storage medium are shown in Figures 10-1 and 10-2. Heat is collected and stored in water, and freezing is avoided by draining the collector when it is not in use (Figure 10-1), or when freezing might occur (Figure 10-2).

A primary requirement of all types of drain-back liquid systems is the design and positioning of collectors and piping on at least a slight slope. Any incidental upturns or lengthy horizontal piping sections result in incomplete drainage. Freezing can then cause rupture of pipes and collector tubing.

Probably the most reliable type, shown in Figure 10-1, requires no specific controls for collector drainage. When pump operation is interrupted by the proper control signal, water drains from the collector back through the idle pump into the tank, while air rises into the collector from the top of the storage tank through the large (one-to two-inch) pipe between collector outlet and storage. If there are no water traps in the collector and piping, and if the down-flow pipe is of adequate size, this system is virtually fail-safe. But the lack of a water-filled, down-flow pipe requires the pump to deliver water against the full static head between collector top and tank water level, and the nightly exposure of the collector tubing and system piping to humid air increases normal liquid corrosion rates.

Pumping power can be minimized by use of the siphon return design shown in Figure 10-2. Static head between storage and the top of the collector is recovered in the water-filled, down-flow return line (typically one-half- to three-fourths-inch diameter). An automated air inlet

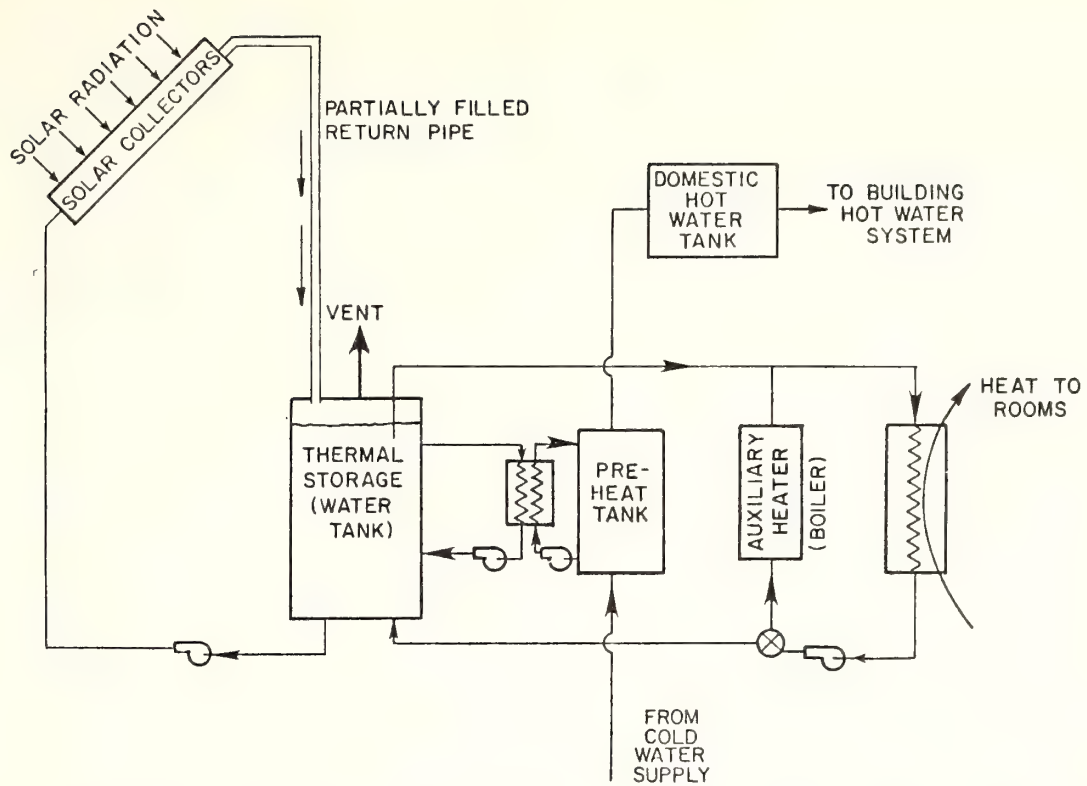


Figure 10-1. Schematic Diagram of a Single-Liquid (Drain-Back) Space Heating and Water Heating System (Gravity Return)

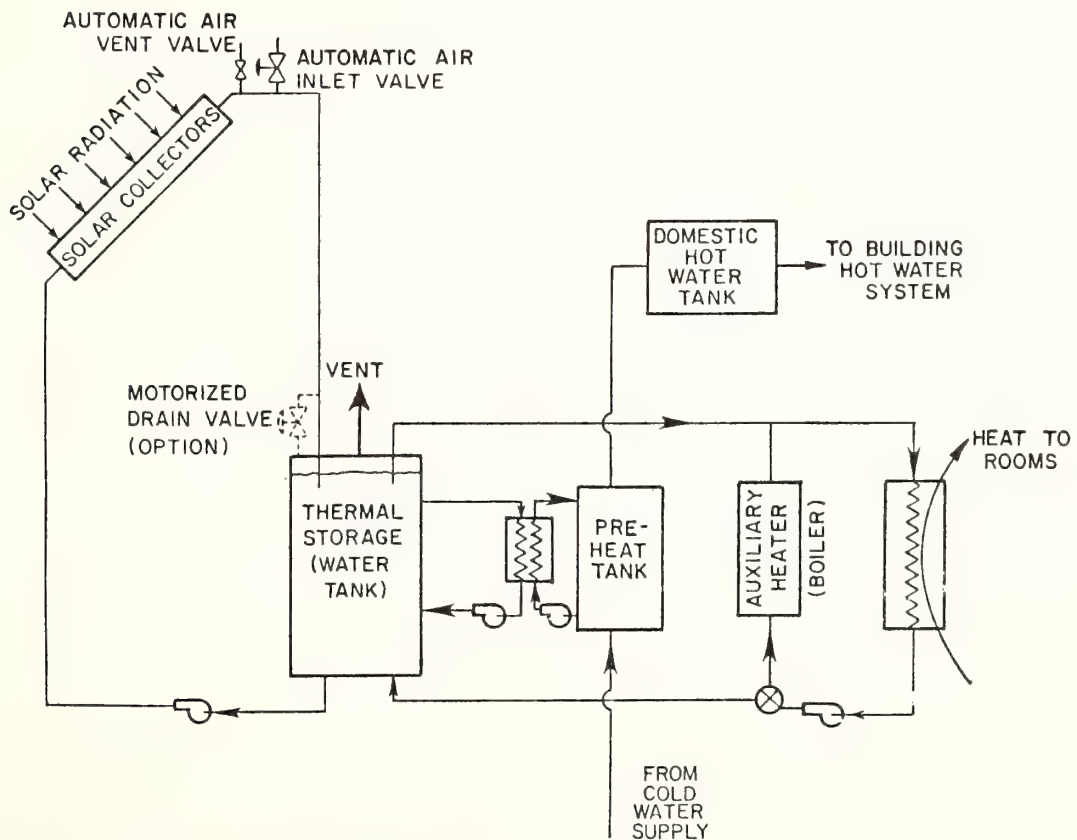


Figure 10-2. Schematic Diagram of a Single-Liquid (Drain-Back) Space Heating and Water Heating System (Siphon Return with Motorized Air Inlet Valve)

valve ("siphon breaker") at the collector top opens when the collector temperature approaches the freezing point (or if there is a power outage). The open air inlet permits drainage of the collector and piping into the storage tank through the two connecting pipes while the collector fills with air from the atmosphere. When collector temperature again rises, pumping resumes and air is forced out of the collector circuit through the automatic air vent valve, which closes when air is no longer being vented. Although this system requires less pumping power than the non-siphoning type shown in Figure 10-1, dependence on a control sensor and automatic valve, even though opening without power, increases the risk of freeze-damage.

Three additional design variations in single-liquid (water) systems are used with sufficient frequency to merit description. The first is a modification of the drain-back design shown in Figure 10-2. Instead of using an electrically actuated air inlet valve, a motorized water drain valve on a branch of the return pipe (shown in dotted outline) may be used. When a sensor in the collector shows a temperature near the freezing point, (or if there is a loss of electric power), this valve opens. Because of the pipe opening into the air space in the tank, the weight of water in the return line causes a negative pressure at the top of the collector. This negative pressure causes the automatic air inlet valve to open and the collector to drain. When pumping is again started, both valves close, air is eliminated through the vent near the top of the collector, and water circulation is reestablished.

Another procedure for draining the collector is used when a pressurized hydronic system is required. Most conventional hot water heating systems operate at a hydraulic pressure of at least 15 psi. When a single-liquid solar system is incorporated, all components are under pressure. The storage tank is completely water-filled and a small expansion tank (sometimes called a compression tank) is used as in conventional hydronic systems. Freeze protection is provided by use of a motorized valve in the collector-storage circuit which opens to a drain (sewer). The pressure decrease causes the automatic air inlet valve to admit air to the collector while a motorized valve on the line from the tank outlet to the pump and a check valve on the line from the collector to the storage tank both close. Pressure is thus maintained in the storage tank as a small volume of water in the collectors and piping is drained away. Make-up water must be automatically supplied to replace the volume discharged. Start-up of the system involves reversal of the operations, air being purged through the automatic vent.

A third alternative design involves circulation of water from storage through the collector when freezing would otherwise occur. In areas where freezing is very infrequent, a low-temperature sensor in the collector causes the pump to supply sufficient flow from storage to prevent freezing. The loss of heat which this procedure entails need not be excessive in regions suitable for this design. This is not a fail-safe design because a power outage or pump failure might occur during freezing weather. This risk can be avoided by supply of service water (from pressurized mains) to the collector through another automatic valve to the drain. Both valves can be actuated (opened) by low-temperature signal or by loss of power.

DUAL-LIQUID COLLECTION SYSTEM

A widely used solar liquid heating design is shown in Figure 10-3. The system is arranged to collect solar heat in a non-freezing liquid, deliver the heat to water storage via a liquid-to-liquid heat exchanger, and supply heat from storage to the space heating and service hot water equipment. When solar heat cannot meet the demand of either domestic water or space heating, auxiliary heat is supplied from one or both of the conventional units shown. Although a solar domestic hot water heater can be completely separate from a solar space heating system, it is more convenient and economical to combine them. During the warm months of the year the collectors in an integrated system, which would otherwise be unused, can supply practically all of the required domestic water heating.

The most commonly used liquid in the collector loop is a mixture of water and ethylene glycol (ordinary automobile radiator antifreeze), although propylene glycol and water may also be used. The advisable glycol concentration depends on the minimum temperature expected in the region where used. A 50-50 mixture provides maximum protection to -34°F . A centrifugal pump, usually at the lowest position in the loop, circulates the liquid solution through collectors and heat exchanger, typically at a rate of about 0.02 gallon per minute per square foot of collector. An expansion tank with open vent or pressure relief valve is installed preferably at the highest point in the loop. This tank should have a sight-glass or other liquid level indicator, and it should have a volume equal at least to half the volume in the collector loop. This tank is also a convenient point for charging the system with liquid.

In many dual-liquid systems, the collector loop is pressurized to a moderate (15 psi) pressure. The likelihood of boiling is thereby reduced, positive pressure at pump inlet is assured, and fluid loss is minimized. Instead of a vented expansion tank near the top of the collector, a pressurized expansion tank of a few gallons capacity (such as in conventional hydronic heating systems) is usually mounted in the line near the circulating pump. This tank contains a flexible bladder which separates the liquid from pressurized air in the upper part of the tank. An automatic air vent and a pressure relief (safety) valve are also provided. After filling the collector loop through a suitable pipe connection on the suction side of the pump, compressed air is supplied to the expansion tank (usually by a hand pump) up to the desired pressure.

In a more limited number of systems, some type of oil is used as the heat collection fluid. Silicones, diphenyl, Dowtherm[®], Therminol[®] are examples. In addition to providing complete protection from freezing, these liquids will not boil at temperatures attainable in flat-plate collectors and they are non-corrosive. They are more expensive than antifreeze solutions, however, and their heat transfer properties are not as favorable.

A heat exchanger and water heat storage are always used in systems with non-freezing liquids because of the excessive cost of filling a thermal storage tank with such expensive liquids. The heat exchanger is usually a commercial tube-and-shell type through which water is circulated from storage by a centrifugal pump at a volumetric rate one to two times that in the collection loop. The pump is supplied from the bottom of the tank, and after being heated in the exchanger, the water is

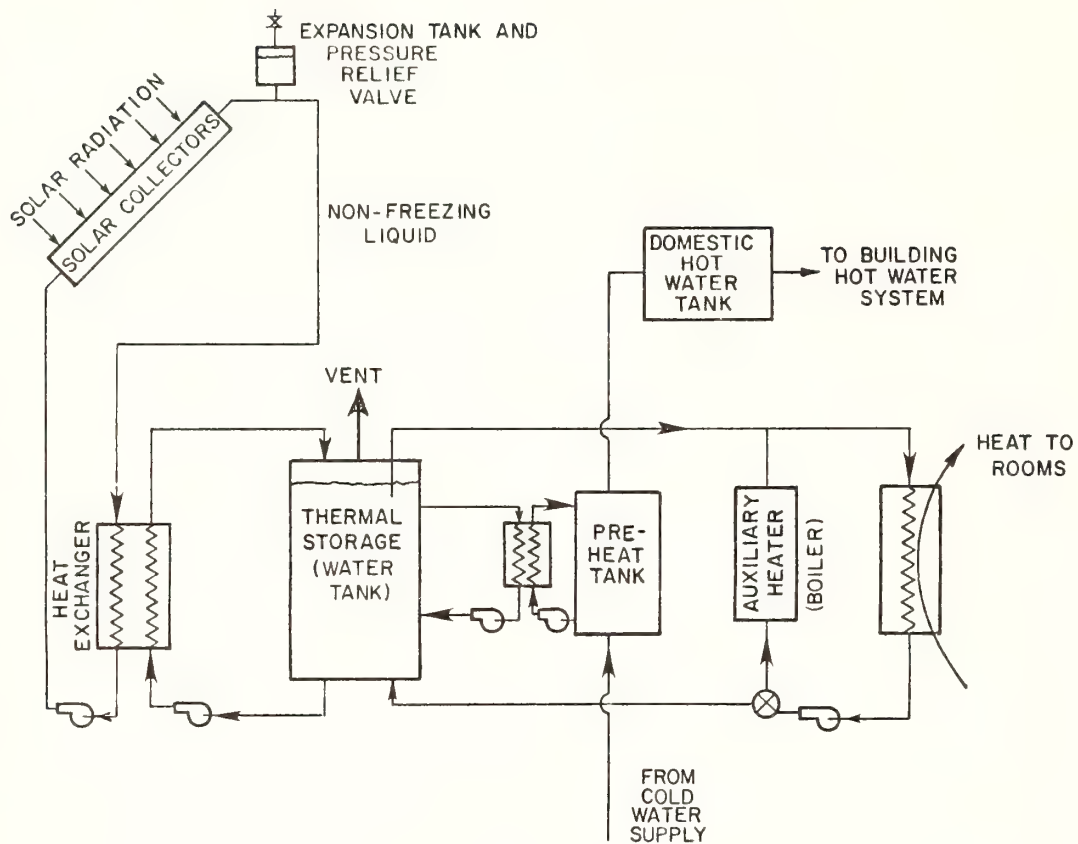


Figure 10-3. Schematic Diagram of Dual-Liquid Space Heating and Hot Water System

returned to the tank near its top. In some systems, storage is heated more directly by circulating the collector fluid through long coils of tubing submerged in the tank. The water pump is thus eliminated, but pressure loss through the coil is higher than through an external heat exchanger, so power requirements in the collector pump are higher and access to the heat exchange surface and its connections is more difficult.

LIQUID SUBSYSTEMS AND COMPONENTS

The individual components of heating systems in which solar heat is collected and stored in liquids are discussed in Module 7. The discussion is continued here, but with emphasis on equipment characteristics which affect, and are affected by, system design and performance.

COLLECTORS AND STORAGE

The efficiency of collectors in a solar system is influenced by all other components in the system. So that heat loss from the collector can be minimized, it should be supplied with liquid at the lowest available temperature. With proper design and operation, moderate temperature stratification in a water tank can be obtained; warm, lower density liquid lies near the surface, and colder, heavier liquid near the bottom of the tank. Water from the bottom of the storage tank should therefore be circulated directly or indirectly to the collector. If a heat exchanger is used, as in Figure 10-3, water is circulated from the bottom of the storage tank, through the heat exchanger, back to the top of the tank. In a drain-down design, as in Figures 10-1 and 10-2, water from the bottom of storage is pumped through the collector. Heated water returns to the top of the tank. Typical rise in temperature of water flowing through the collector or through the heat exchanger is 10°F to 20°F during sunny mid-day periods.

In addition to dependence on heat input from the collector, storage temperatures are affected by the rate of heat delivery to the load, dependent, in turn, on the characteristics of the load heat exchangers.

Selection and sizing of heat exchangers are therefore important in system design. While oversizing the heat exchangers has minor influence on system performance, undersizing can reduce the quantity and efficiency of solar heat collection.

COLLECTOR-STORAGE HEAT EXCHANGERS

If a non-freezing liquid is circulated through the collector, a heat exchanger must be provided to transfer heat from the collector fluid to water storage. Because of temperature limitations in flat-plate solar collectors, the temperature difference across heat exchangers should be small. This temperature difference is minimized in two ways: (1) by providing a large surface area for heat transfer in the exchanger and (2) by maintaining high flow rates through the exchanger. Tube-in-shell heat exchangers are simple, efficient, and readily available. They usually consist of multiple tubes enclosed within an outer shell. One fluid passes through the tubes while the other fluid passes outside the tubes. Large heat transfer surface can be provided in compact arrangements.

The performance characteristics of a single-pass, counterflow heat exchanger are illustrated in Figure 10-4. A single tube is shown for simplicity but multiple tubes are usually involved. It can be seen from the temperature profiles that the temperature difference between fluids is reasonably small along the length of the heat exchanger.

The manufacturer's guide should be followed in selecting a heat exchanger. If appropriate information is lacking, the manufacturer's representative should be consulted for assistance and/or advice. Data necessary for heat exchanger sizing and fluid flow rate determination

are the temperatures of the two fluids entering the exchanger and the Btu-per-hour heat transfer rate desired.

At high fluid velocities and flow rates, good heat exchanger effectiveness can be achieved, but at the expense of pumping power. A practical compromise in these two opposing objectives is sought in system design.

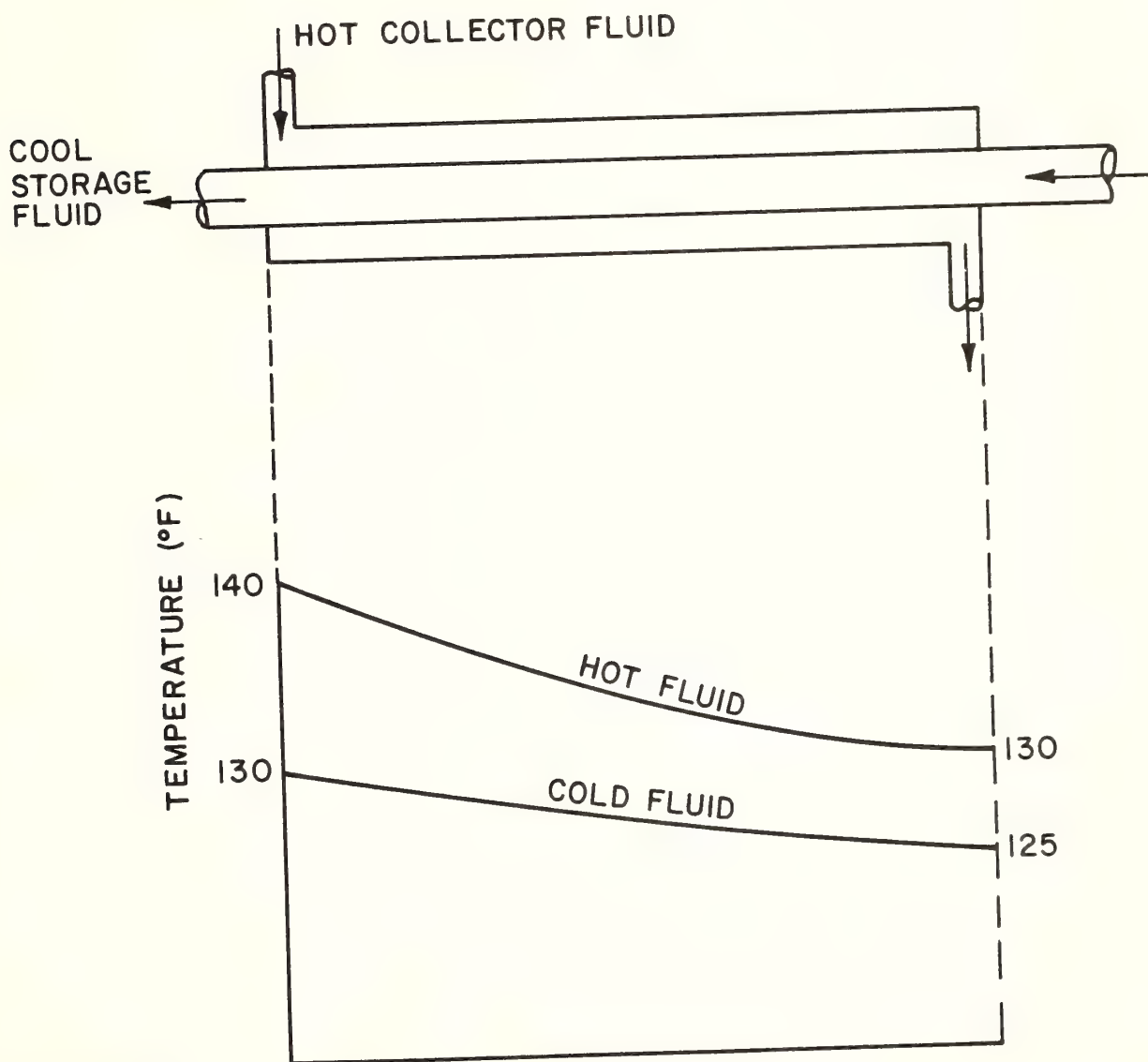


Figure 10-4. Typical Temperature Profiles in a Single-Pass Counterflow Heat Exchanger

SUBSYSTEMS FOR SUPPLY OF SOLAR HEAT TO USE

Heat from a solar hot water storage tank may be distributed to living spaces either in hot water or in warm air. For water distribution, methods and equipment commonly used in conventional hydronic heating systems are employed. When activated by a room thermostat, a pump supplies solar-heated water from the top of the storage tank to one of several types of heating coils in each heated space in the building and returns the water to the lower part of the tank. Figures 10-1, 10-2 and 10-3 illustrate this application, the schematic heating coil in the building representing any type of liquid-to-air exchanger.

Conventional baseboard strip heaters widely used with fuel-fired and electric hot water boilers* normally operate at water supply temperatures of 160°F to 180°F. The limited heat transfer surfaces formed by a single water tube through closely spaced metal fins necessitate the use of these comparatively high temperatures. Temperatures in storage tanks supplied from typical flat-plate collectors seldom exceed 150°F in winter, so either additional baseboard heating surface is needed in solar systems or solar heat usage in cold weather must be heavily supplemented. A doubling of the usual baseboard heat transfer surface permits reduction in design (maximum) temperature to 130°F to 140°F and a substantial increase in solar heat usage at these temperatures where collection efficiency is materially improved.

Solar-heated water can be distributed in conventional hydronic systems involving multiple fan-coil exchangers, pipe coils imbedded in floors and ceilings, and cast iron hot water "radiators". Temperatures

* So-called boilers for residential space heating may in fact produce steam, but more commonly they serve as water heaters for hydronic (liquid water) heat distribution systems.

and flow rates are compatible with the requirements of most fuel-operated systems.

In ducted warm air heating systems, hot water is supplied from solar storage to a finned coil heat exchanger in the main air duct and returned by a centrifugal pump to the tank (Figure 10-5). Air is circulated through the exchanger by a conventional fan or blower, and is usually heated from about 70°F to a temperature within 10°F to 15°F of the hot water supplied from storage. As explained below, a conventional warm air furnace is usually employed in this system so that the temperature of air supplied from the solar coil can be increased when needed.

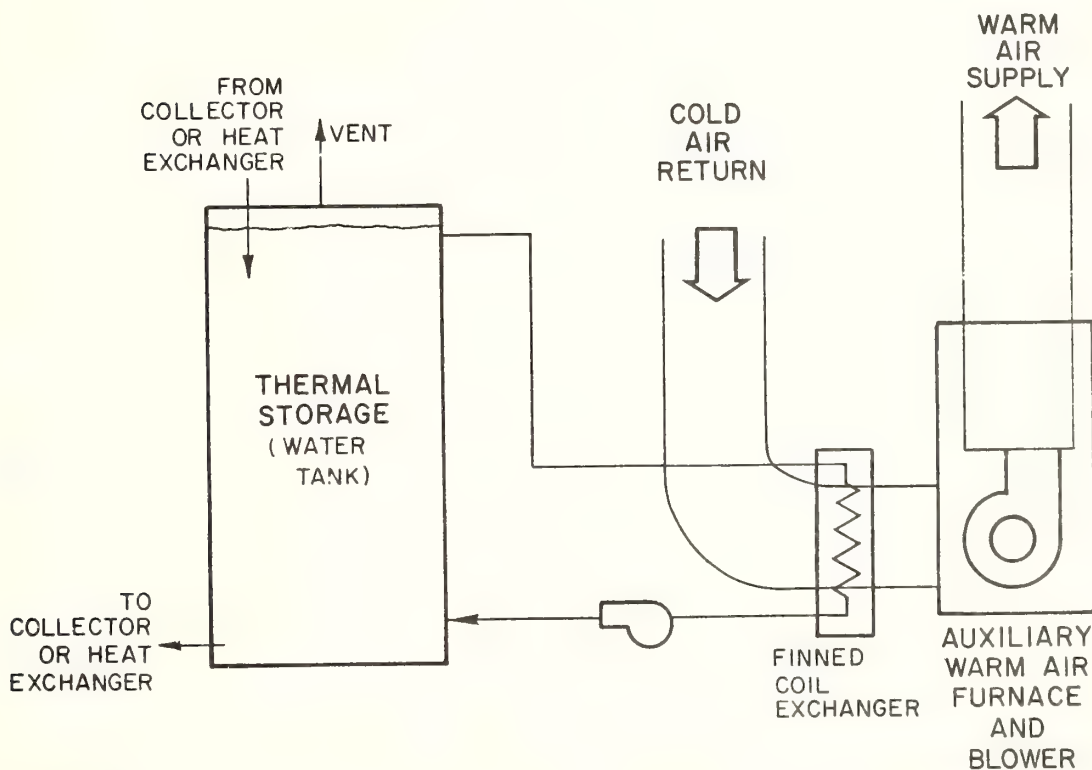


Figure 10-5. Solar Heating with Auxiliary Furnace

AUXILIARY HEAT

There are several methods for supplying auxiliary heat in a liquid solar system for space heating. In virtually all practical solar heating designs, auxiliary heat is supplied to the fluid stream in which heat is distributed to the various zones in the building. With hydronic distribution (baseboard strips, individual fan coils, imbedded tubing, cast radiators), a fuel-fired or electrically heated hot water boiler is used as in Figures 10-1, 10-2, and 10-3. If a central heat exchanger and ducted warm air are employed, as in Figure 10-5, auxiliary heat is most economically and practically supplied in a furnace or electric heater through which the solar-heated air passes to the rooms.

In hydronic distribution systems, the hot water boiler is best used in parallel with the solar supply from storage rather than in series with it, so that only one source is used at a time. The series design is seldom used because some of the heat supplied to the water passing through a thermostatted auxiliary boiler would flow on through the load exchangers and be accumulated in the solar heat storage tank. The resulting temperature rise in solar storage would reduce collector efficiency and capacity for solar heat storage. Figures 10-1, 10-2, and 10-3 show a single pump and automatic valve for hot water supply either from solar storage or from the auxiliary boiler. Suitable control equipment, explained below, regulates the system so that when the demand cannot be met by solar, auxiliary is used.

If heat is distributed in warm air heated by exchange with solar-heated water, auxiliary heat is usually supplied to the air in a warm-air furnace located downstream from the solar coil as shown in Figure 10-5. There is no possibility that auxiliary heat can affect

solar storage in this design, so the series arrangement is advantageous. Use of stored solar heat even at comparatively low temperature (e.g. 80°F to 90°F) is thus made possible by further heating of the tepid air to useful temperatures of 130°F to 160°F.

AUXILIARY HEAT PUMP

A special type of auxiliary heater for solar systems is an electrically driven heat pump. A heat pump uses electrical energy to extract heat from a low temperature source and deliver the heat at a higher temperature. By this process, heat delivery at useful temperature may be several times the electric energy input. In practice, annual performance factors (heat delivery per unit electric input) of about 2 are commonly achieved, although in colder climates, a somewhat lower factor generally prevails. The process is identical to a refrigeration cycle, and the same machine that is used as a heat pump in winter may be used as a refrigeration air-conditioner in summer. Switching between heating and cooling is usually done inside the machine by reversing the evaporator and condenser units, as shown in Figures 10-6 and 10-7.

One method of heat pump use in a warm air distribution system, illustrated in Figure 10-8, involves the condenser coil (heating coil) of an air-to-air heat pump in the air circuit downstream from the solar-to-air exchanger. Because of pressure limitations in the heat pump, the flow of water through the solar exchanger is usually interrupted when auxiliary heat is required so that air is not supplied to the heat pump evaporator coil at excessive temperatures (e.g., not above 100°F). As in conventional heat pump installations, electric resistance coils are provided for use during severe cold weather.

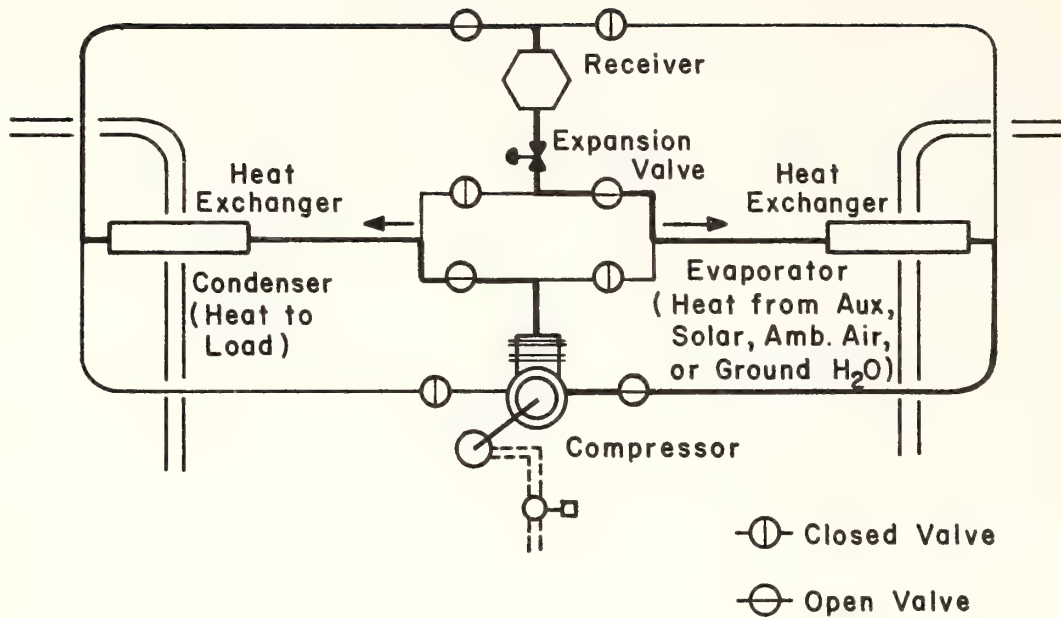


Figure 10-6. Heat Pump in Heating Mode

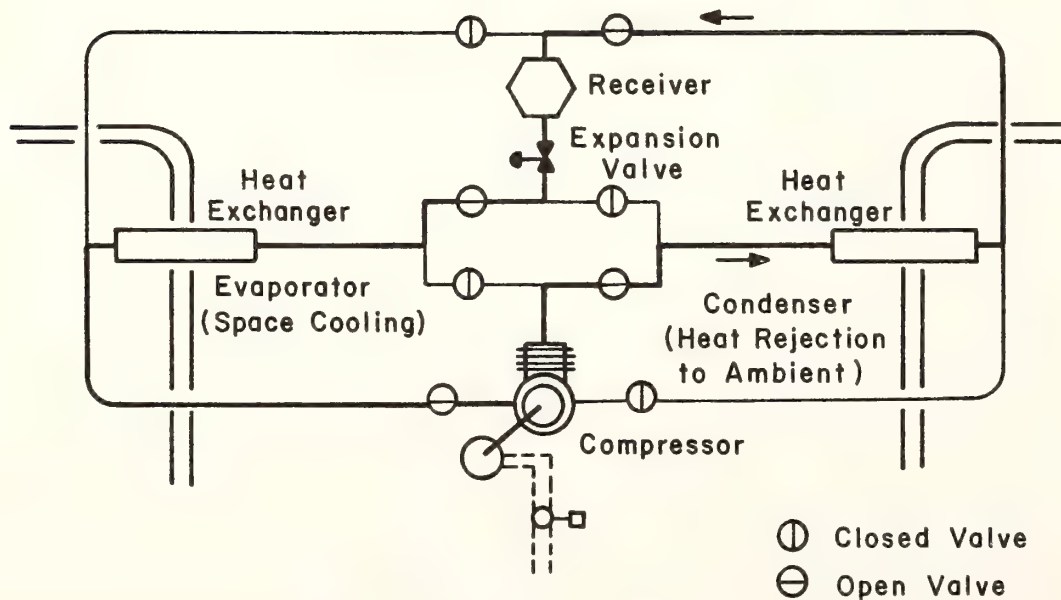


Figure 10-7. Heat Pump in Cooling Mode

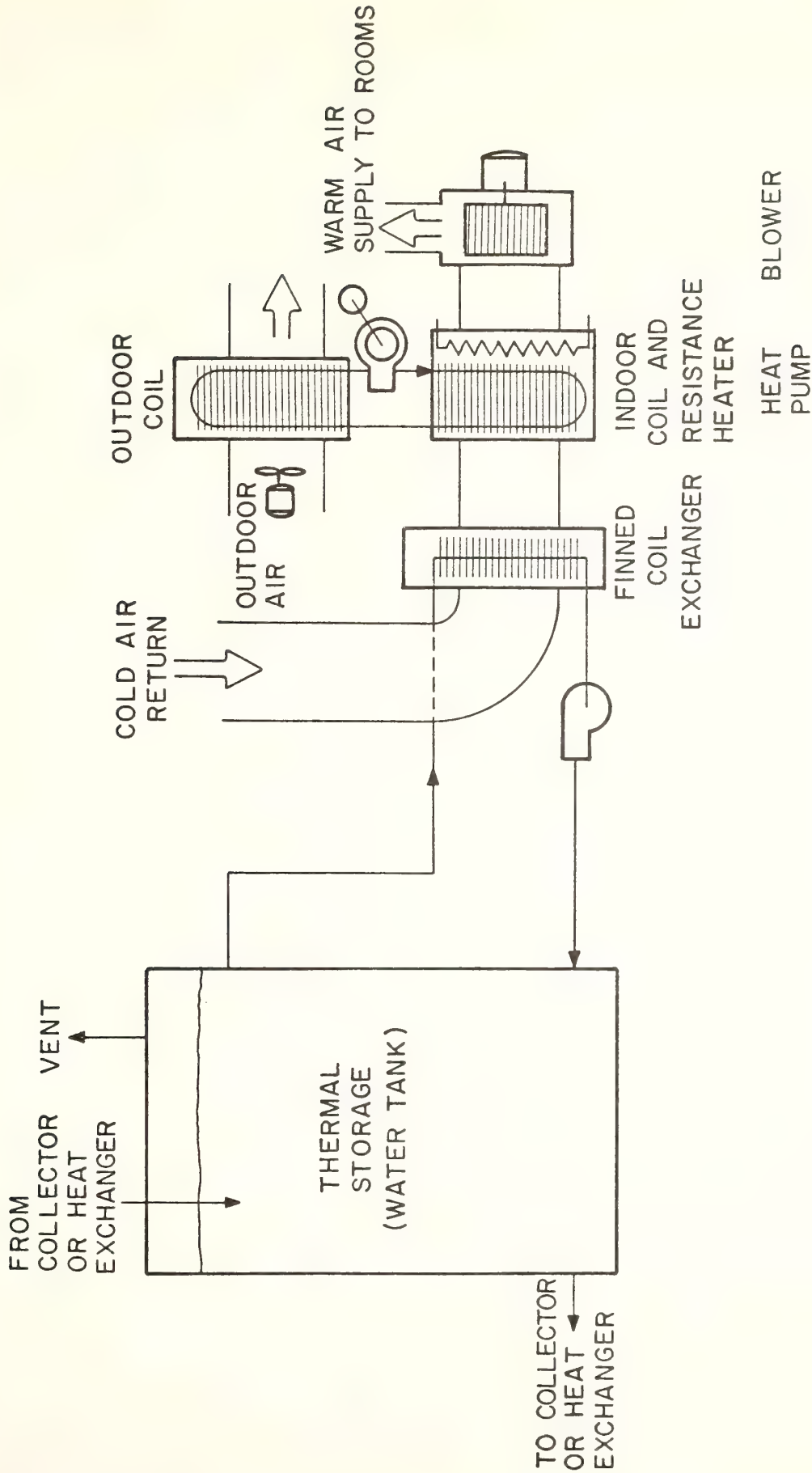


Figure 10-8. Solar Heating with Auxiliary Air-to-Air Heat Pump

Another design (Figure 10-9) involves an air-to-water heat pump rather than a hot water boiler for auxiliary heat supply. Replacement of the boiler in Figure 10-1, 10-2, or 10-3 with the heat pump condenser coil and back-up electric resistance heater permits a reduction in energy consumption by virtue of a COP greater than unity. Heat distribution may be in water or air as with fuel auxiliary.

A third method for combining a heat pump with a solar heat supply is shown in Figure 10-10. In this application the water-to-water type is often referred to as a solar-assisted heat pump. The concept of a solar-assisted heat pump is the supply of stored solar heat to the evaporator of the machine at a temperature higher than outdoor ambient air. Higher COP and lower collector supply temperatures (with higher efficiency) could then result. If water in solar storage is not hot enough to meet demands directly, it is used as a source of heat to the heat pump evaporator coil. Heat is then supplied to the building by heating water in the condenser coil, with circulation to the living space or by exchange with air. In multizone distribution systems where individual water-to-air heat pumps in each zone are supplied with low-temperature water, the use of solar preheated water as the heat pump supply may reduce electricity use, provide heated air where and when needed, and minimize heat distribution cost. Because the fluid temperature delivered from the collectors may be as low as 80°F to 100°F, collection efficiency can be higher than that obtained in a solar system supplying heat directly to use at water temperatures of 120°F to 180°F. It might also be possible to use cheaper collectors designed specifically for operation at lower temperatures. When solar storage may approach the freezing point in midwinter, as a result of large withdrawals of

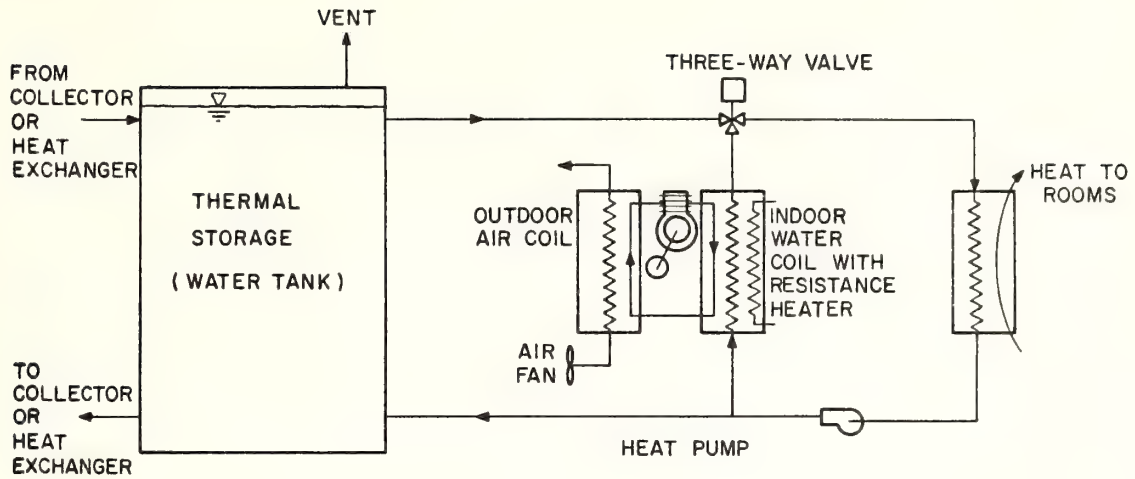


Figure 10-9. Solar Heating System with Auxiliary Heat from Air-to-Water Heat Pump

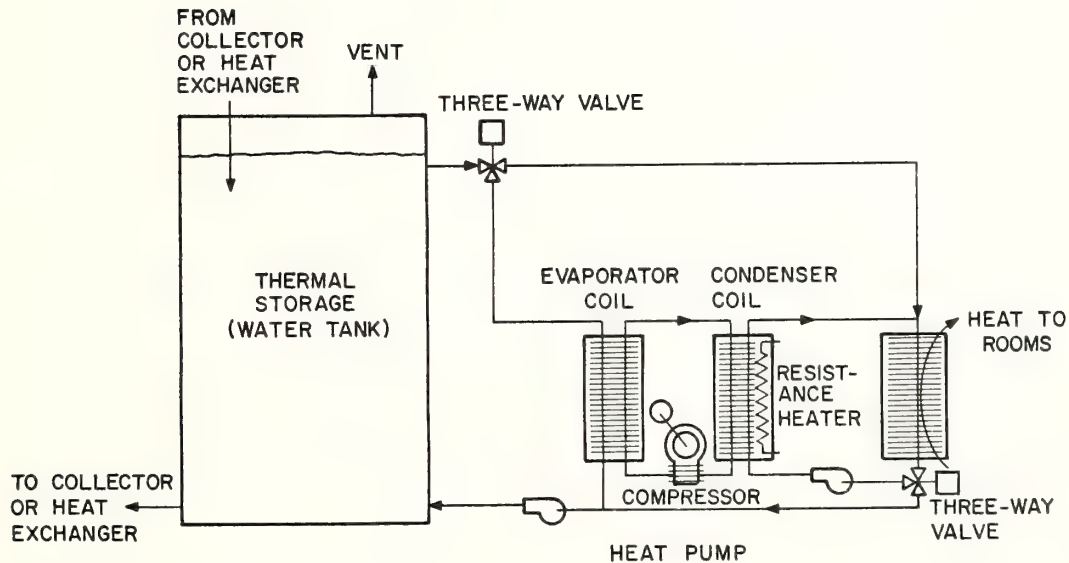


Figure 10-10. Solar-Assisted Heat Pump (Liquid-to-Liquid) Heat Pump in Series with Solar Storage

heat for heat pump supply, electric resistance elements are called upon to meet the demand.

This series arrangement of collector to storage to heat pump to load is handicapped, however, by excessive draw-down of storage in midwinter so that storage temperature is rarely sufficient for direct heating of the building. Nearly all of the heating demand during the cold months must therefore be met by electric energy either as heat pump supply or as resistance heating. The annual electricity requirement has usually been found greater than that for operating the parallel systems in which ambient air is the heat pump source.

SERVICE HOT WATER

Solar heat for service hot water is usually obtained by circulating water from the top of the main storage tank through a double-walled heat exchanger as shown in Figures 10-1 to 10-3. Simultaneously, potable water from the solar pre-heat tank is circulated through the heat exchanger and back to the top of the pre-heat tank. These pumps operate whenever the temperature in the storage tank is greater than the water temperature in the pre-heat tank by a preset amount. When useful heat cannot be delivered from storage to the pre-heat tank or when the pre-heat tank has reached a limiting high temperature, say 175°F, the pumps do not operate.

During the heating season, water temperature will frequently be less than 140°F, so dependable delivery of hot service water requires an auxiliary heater. Solar heat is therefore used to pre-heat cold water from the water main before it enters the hot water heater. During the summer, the water temperature in the pre-heat tank will usually be greater than 150°F, so auxiliary heating is only occasionally required.

Suppose that average use of service hot water in a household is 75 gallons per day. Also assume that the water temperature from the main is about 60°F and that the desired water delivery temperature is 140°F. The daily quantity of heat necessary to raise the temperature of the service water from 60°F to 140°F is therefore about 50,000 Btu. Delivery of 50,000 Btu from 1000 gallons of storage to the service water heating system will cause a drop in storage water temperature of 6°F (assuming no heat is delivered from the collectors to storage in this period). If the storage tank temperature is less than 140°F, useful heat delivery to the service water heating system will be less than that indicated above, and the auxiliary heating unit will be required to maintain the desired water temperature in the hot water heater. In the summer months, there is usually enough heat in the solar heated tank to supply all of the service hot water requirements.

To comply with most plumbing codes, the heat exchanger between main solar storage and the potable water supply must be double-walled. By this design, a perforation in a tube wall will not permit mixing of the two fluids, regardless of pressure difference. The resulting water leak will, instead, run out of the exchanger and prevent cross-contamination. Several designs for accomplishing this purpose are available.

Another option for auxiliary use is an electric resistance heating element near the top of the solar pre-heat tank rather than a separate conventional water heater. Solar heated water is brought into the tank at a level below the electric heating element so that temperature stratification can be maintained. This design accomplishes the same purpose at somewhat lower cost than incurred with a separate electric water heater as auxiliary.

PUMPS, PIPING AND ACCESSORIES

Centrifugal pumps are normally used for circulating liquids in the several loops of a solar heating system. For the design flow rates and system head losses, pumps may be selected from stock items in catalogs, or made to specifications by pump manufacturers. Centrifugal pumps should be located so that priming is not necessary, five feet of head on the suction side normally being sufficient.

A check valve, strainer or filter, and an expansion tank should be installed in the collector loop. Figure 10-11 shows a portion of a vented, dual-liquid system. The check valve prevents thermosiphon flow and heat loss from the collector when the pump is not running and the collector is colder than the storage tank. A filter or strainer prevents entry into the collector of any particulate matter in the piping and collector fluid.

An expansion tank must be provided in the collector loop to accommodate the increase in volume of the collector fluid as it is heated. The volume of the tank should be at least half the volume of the fluid in the collectors and headers. If positioned near the top of the collector array, it may be vented to the atmosphere or provided with a pressure relief valve to prevent dangerous pressure increase in the collector loop if boiling occurs. Alternatively, an expansion tank containing a flexible diaphragm or bladder may be used anywhere in the collector loop, and a pressurized system may then be operated.

In single-liquid, drain-back systems, pump and piping considerations have been outlined in an earlier section of this module.

The amount of electric energy used for circulating liquids through collectors, heat exchangers, storage tanks, and distribution systems

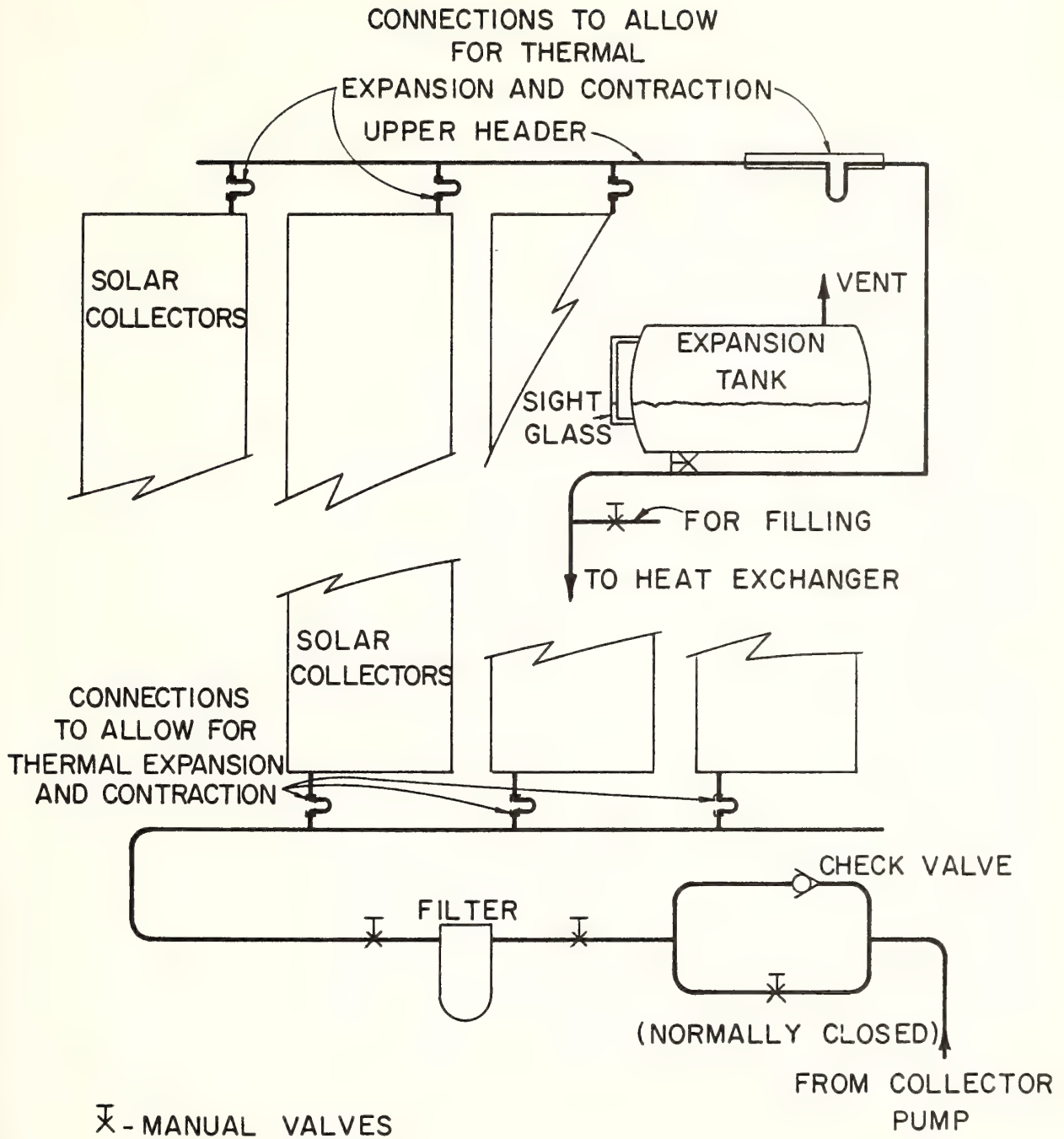


Figure 10-11. Collector Loop Fittings and Details, Dual-Liquid System

depends heavily on system design and collector type. In systems designed to require minimum pumping power, electricity use can be as low as five percent of solar heat delivery, but ten to fifteen percent is commonly employed.

Piping may consist of either copper or high temperature (CPVC) plastic pipe, and all pipes should be insulated with appropriate material such as fiberglass pipe insulation at least one inch in thickness. Care should be taken to allow thermal expansion of the pipes, and long pipe lengths should provide more freedom for expansion than short lengths. Pipes should be sized so that water velocity does not exceed five feet per second. In Table 10-1, recommended pipe diameters are indicated for various flow rates.

Table 10-1

Recommended Pipe Diameters for Various Flow Rates

Select Pipe Size	Gallons per Minute	Velocity FPS	Pressure Drop per 100 feet, PSI
3/8	2	3.36	6.58
1/2	4	4.22	7.42
3/4	8	4.81	6.60
1	15	5.57	6.36
1-1/4	25	5.37	4.22

SYSTEM INTEGRATION AND CONTROL

Assembly of the several solar components into a well-functioning system, and control, are requirements of exceptional importance. Collector, storage, heat exchangers, auxiliary heater, pumps and piping,

and control system must be mutually compatible in size, function, materials, reliability, and cost. Although a great variety of components may be assembled into functional systems, experience shows that certain equipment combinations and control methods provide superior performance.

Previous subsections of this module in which solar collectors and heat storage were discussed show the principal methods for combining these two major components. Design studies and practical experience show that the storage capacity for the most economical heat supply is sufficient for accumulating a full day's solar collector output, and that the cost of larger storage is greater than the value of the small added increment of solar heating capability. Approximately 1.5 to 2.5 gal/ft² of collector are therefore usually employed, with 1.5 a practical average. Even though the storage tank is usually inside the building, it should be well insulated with fiberglass or plastic foam having an insulating value of at least R-10.

The relative merits of drain-back and dual-liquid collector/storage systems have been explained. Pumps, valves and piping in either system should be sized in accordance with flow requirements, manufacturers' specifications, and accepted practice. All components should be placed in locations where there is convenient access for maintenance. Suitably oriented roofs usually provide the most economical base for supporting the collector, and the mass of the storage unit nearly always dictates its support on a basement floor or a ground floor. The tank can be installed in the ground near the building, but limited access, moisture problems, and heat loss in buried installations make storage in the heated space a definite advantage. The other system components can be advantageously located near the tank.

Selection of collector size (which also results in storage size determination) is a matter of balancing numerous design factors. Climate (annual heating degree-days), the heat loss rate and hot water demands of the building, solar energy availability, the fraction of total heat requirements to be met by solar, the solar collector characteristics, and the control and configuration of the complete solar system, all affect the required collector/storage size. Several procedures of various complexity and accuracy for collector sizing are explained in Modules 6 and 11.

SYSTEM CONTROL

The fundamental operating modes of solar heating systems are collecting and storing solar heat and delivering heat from storage to load. The auxiliary unit must be controlled to provide heating when there is insufficient solar heat in storage to meet the demand. A typical control system is shown schematically in Figure 10-12, and the functioning of its principal elements is shown in Table 10-2.

A practical method for controlling the solar collection process involves use of temperature sensors which actuate the collector pump (and also the storage circulation pump if a heat exchanger is used) whenever the temperature of the liquid leaving the collector exceeds the lowest temperature in storage by a preset amount, say 15°F* (first line in Table 10-2). Sensor S1, placed in the flow passage as close as possible to the collector exit, and sensor S2, near the bottom of the storage tank where storage water is likely to be coldest, provide the signal which is compared electronically with a signal produced by a preset

* This difference must include a 5° to 10° difference in the heat exchanger, if a dual-liquid system is used.

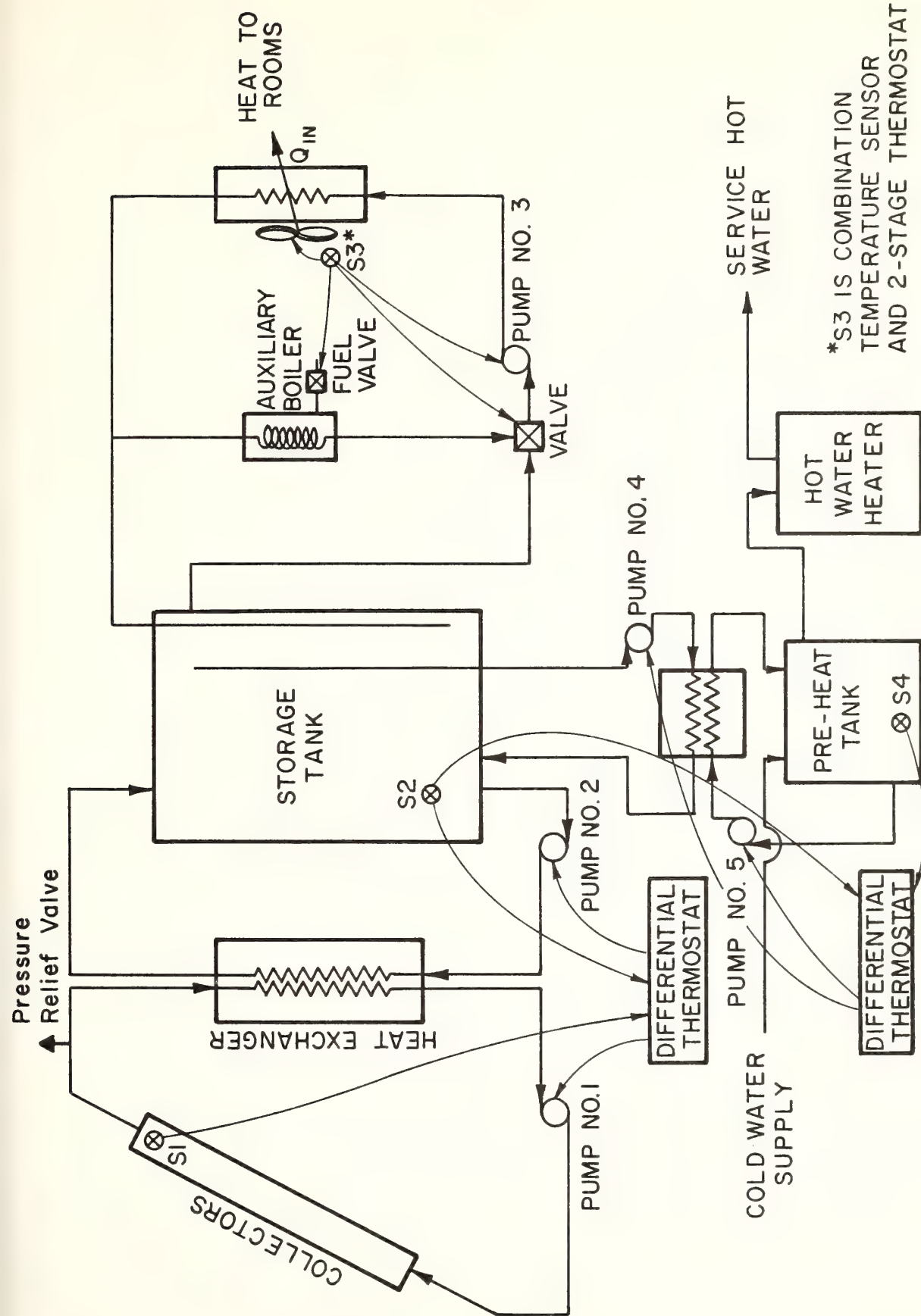


Figure 10-12. Typical Controls for Dual-Liquid Solar Heating System

Table 10-2

Control Truth Table for Dual-Liquid Solar Heating System
(in Figure 10-12)

OPERATION	CONDITIONS				MOTORS							
	S ₁	S ₂	S ₃	S ₄	P ₁	P ₂	P ₃	P ₄	P ₅	Water Valve	Fuel Valve	Fan
Collecting Solar Heat	$T_1 > (T_2 + 15)$	-	-	-	on	on	-	-	-	-	-	-
Heating Service Hot Water	-	-	-	$T_4 < (T_2 + 20)$	-	-	-	on	on	-	-	-
Space Heating with Solar	-	$T_2 > 90$	$T_R < (ST_3 - 1)$	-	-	-	on	-	-	Open to Storage	Closed	on
Space Heating with Fuel	-	-	$T_R < (ST_3 - 3)$	-	-	-	on	-	-	Open to Boiler	Open	on
Space Heating with Fuel	-	$T_2 < 90$	$T_R < (ST_3 - 1)$	-	-	-	on	-	-	Open to Boiler	Open	on

NOTES:

- = not relevant

T₁ = temperature at S₁T₂ = temperature at S₂T_R = room temperature at S₃ST₃ = room thermostat setting (e.g. 70°F)T₄ = temperature at S₄

Temperature differences shown (1,3,15, and 20 degrees) may be considered typical factory settings, usually adjustable by installer. Setting for overriding first stage heating (from solar tank), shown as 90 degrees, also typical and adjustable by installer.

temperature difference to start and stop circulation pumps No. 1 and No. 2. The temperature difference which shuts off the pumps, say 3°F, is less than the temperature difference which turns them on, so that excessive on-and-off cycling at the beginning and end of the day is avoided.

When liquid first circulates through the collectors in the morning, the temperature rise from inlet to exit is only a few degrees (typically 3° to 5°F) because of low solar intensity. As solar intensity increases, the temperature rise in the circulating liquid increases to 15°F to 20°F by mid-day. During the circulation period, the storage water temperature increases gradually until late afternoon. When the solar intensity decreases to a level such that the collector can no longer provide useful heat to storage, circulation stops.

The best control strategy for heat delivery to the building is by use of a room thermostat, S3, with dual set points. When heat is required, the first stage contact of the thermostat is completed and water from the storage tank is circulated to the load heat exchanger (third line in table). If the storage water is warm enough, room temperature rises, and circulation stops.

If the storage water is not warm enough to deliver heat at a rate greater than the rate of heat loss from the building, room temperature continues to fall until the second thermostat contact is made (fourth line in table). The auxiliary water boiler or auxiliary warm air furnace is then activated to restore the rooms to the comfort temperature set at the thermostat. If an auxiliary water boiler is used, an automatic valve is also positioned to terminate water flow from the storage tank and to supply hot water from the auxiliary unit to the load heat exchanger. When room temperature rises to the preset temperature, the

entire heat supply system then shuts off. In most designs, a low temperature sensor in storage, S2, can override the above sequence so that if storage is colder than a preset temperature, say 90°, the first contact in the room thermostat causes immediate use of auxiliary rather than the inadequate solar storage (last line in table).

The control of a warm air furnace auxiliary is also by a second (lower temperature) contact in the room thermostat. When it is actuated, energy (fuel or electricity) is supplied to the furnace, all other operations continuing. Thus, the solar supply via the load-exchanger is augmented by auxiliary heat. If, however, storage is below a preset temperature, its usefulness is not usually enough to justify operation of the load pump and sensor S2 turns it off. And as in the system involving a hot water boiler, the furnace is switched on immediately by the first contact in the room thermostat if storage is colder than the low-limit setting.

Solar pre-heating of domestic water is best regulated by a temperature difference comparator similar to the control of the collection loop. The temperature sensor S2 at the bottom of the solar storage tank may also be used for the DHW pre-heater loop control, and a typical difference in temperature between the bottom and top of the main solar storage tank may be included in the temperature difference setting. Alternatively, a sensor may be installed at the top of the solar storage tank to control the pre-heater circulation pumps. The difference in temperature between sensors S2 and S4 controls pumps 4 and 5, as shown in Figure 10-12 and Table 10-2.

PROTECTION AGAINST FREEZING AND BOILING

Protection of liquid collector/storage systems against damage caused by freezing or boiling of the collector fluid is a matter both of design and control. Freeze protection in dual-liquid systems requires maintenance of adequate concentration of ethylene glycol in the circulating solution. Periodic checking of the solution both for glycol concentration (by simple hydrometer similar to a tester for automobile radiator protection) and for pH (acidity) by litmus paper or other color indicator is essential to safe long-term operation. Antifreeze tables are commonly available for making up solutions that can provide freeze protection to temperatures ten degrees colder than recorded minimums. A 50-50 solution has the lowest freezing point, approximately -34°F . Because of deterioration and dilution by vapor loss and subsequent make-up with water, glycol concentrations should be maintained well above the lowest requirements.

Decomposition of glycols by heat, with resulting acid formation, poses substantial corrosion hazards unless properly dealt with. Periodic replacement of collector fluid, at one- to two-year intervals, is highly desirable. An occasional pH check can show whether more or less frequent replacement is necessary. The use of neutralizing additives, as needed, can reduce the frequency of solution replacement, but careful monitoring is necessary.

In drain-back systems controlled by sensing dangerously low collector temperature, thereby actuating suitable valves, inspection and checking at the start of each heating season is essential. Piping and manual valves must be free to drain completely.

A design and operating technique for avoiding collector freezing in mild winter climates involves use of water (without additives) in the

collector, but without drainage even when freezing weather occurs. The infrequency of such occasions justifies circulation, preferably at low rate, of water from storage through the collector during such cold, sunless periods. Moderate heat loss occurs, but freezing is avoided at modest cost. Such systems are vulnerable, of course, to equipment failure and power outages and could logically be used only where sub-freezing temperatures are rarely encountered. Another option in such climates is the use of a low temperature sensor to open valves (requiring power to close) that allow a slow flow of water from the main, through the collector, to a drain.

There are several designs and operating procedures for control of boiling in the collector when heat demand is so low that high storage temperatures result. In drain-down and drain-back systems, boiling can be permitted, the vented steam being made up by automatic or manual addition of water to the storage tank. The collector may, alternatively, be allowed to drain when the collector temperature exceeds a limit setting, but prolonged exposure of the collector to the resulting high temperature can reduce its useful life.

Closed loop collector systems (dual-liquid types) may also discharge excess heat by venting steam from the storage tank. The higher boiling point of the antifreeze solution protects it from boiling as long as there is circulation to the collector/storage heat exchanger. Water make-up to storage must then be provided. If circulation is interrupted by pump failure or power outage, boiling of the collector liquid can occur, even to the point of insufficient liquid in the system for subsequent start-up and operation. Water and anti-freeze solution must then be promptly added to the collector loop.

Another method for excess heat disposal in single- and dual-liquid systems involves the use of an air-cooled fan-coil exchanger (similar to an automobile radiator) through which overheated storage water is pumped and cooled. A tank sensor actuates the load pump, the exchanger fan, and a diverter valve when the tank temperature exceeds a preset limit such as 200°F.

In all of the designs for freeze and boiling protection, the most important consideration is high reliability, because even one failure can result in serious equipment damage and high repair cost.

AIR SYSTEMS

In addition to the obvious differences between heat collection in liquids and in air, the following technical and operational factors may be noted.

1. Solar air systems involve the same medium for solar collection and space heating; solar-heated air can be delivered directly to the building without heat exchange or storage.
2. Heat storage can be accomplished in a bed of loose solids, typically 1- to 2-in gravel, which also serves as the heat exchanger.
3. Temperature stratification in a pebble bed and the return of air to the collector directly from the living space both provide low temperature (70°F) air to the collector with resulting favorable efficiency.
4. The combination of air density, specific heat, and practical flow rates provides a considerably higher temperature rise through the collector, typically a 60° to 90° increase, than in a liquid type.

DOUBLE-BLOWER DESIGN

As with liquid-heating solar systems, there are numerous options for integrating a solar air collector and a pebble-bed heat storage unit into a complete building heating assembly. A widely used solar air system requires two blowers, one for circulating air through the collector and the other for supplying warm air to the rooms. A schematic design of a two-blower, air-heating solar system for both space and domestic water heating is shown in Figure 10-13. There are six principal components: solar collector, heat storage unit, air handler, auxiliary heater and fan, a water heating coil, and a controller (not shown). The water heating coil is coupled to a storage tank and circulating pump, and piped to a conventional hot water heater. By combining the blower, dampers, and hot water coil in an "air handler", the installation and operation of the system are simplified. The control sequences to operate the system in all modes are detailed in a "truth table" (Table 10-3). The operating modes are shown in Figures 10-14 through 10-17. In the table and figures, the abbreviation MD denotes a motorized damper and BD a back-draft damper which swings shut except when air is being forced against one of its faces.

Storing Solar Heat and Heating Hot Water

Solar collection and delivery of heat to storage are achieved by circulating air between collector and storage whenever a sufficient temperature rise can be achieved in the collector. The collector blower in the air handler is actuated by a differential thermostat, with the hot sensor in the air passage at the collector exit and the cold sensor near the cold end of the storage bed or in the air passage leading to

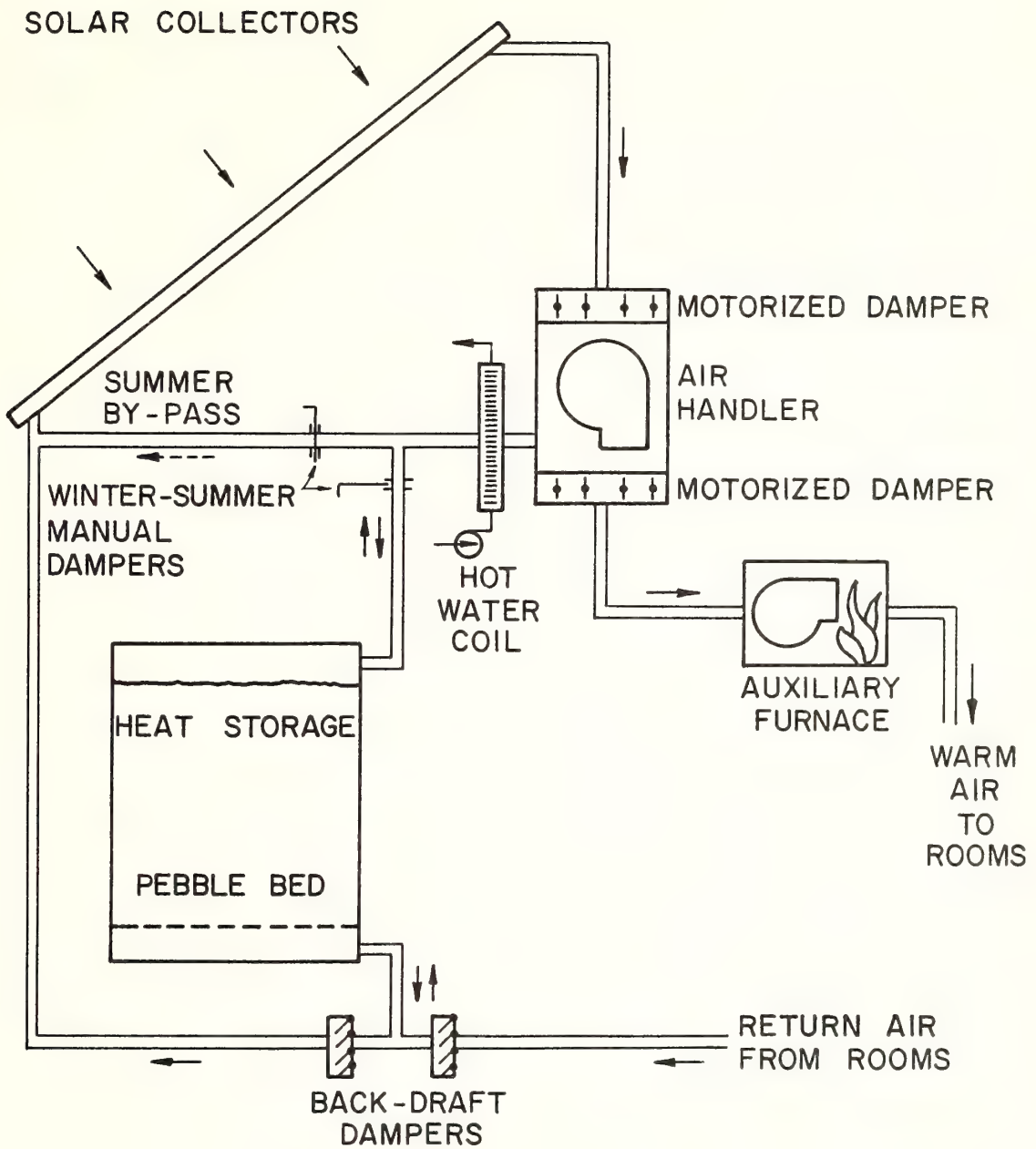


Figure 10-13. Two-Blower Air-Heating Solar System

Table 10-3

Control Truth Table for a Two-Blower, Air-Heating Solar System Operation

Mode	MD 1	MD 2	BD 1	BD 2	Collector Blower	Distribution Blower	Auxiliary Energy	Hot Water Pump	Summer By-Pass
Storing Heat* (Figure 10-14)	Open	Closed	Open	Closed	On	Off	Off	On*	Closed
Room Heating from Collector (Figure 10-15)	Open	Open	Open	Open	On	On	Off	Off	Closed
Room Heating from Storage (Figure 10-16)	Closed	Open	Closed	Open	Off	On	Off	Off	Closed
Room Heating from Collector and Auxiliary (Figure 10-15)	Open	Open	Open	Open	On	On	On	Off	Closed
Room Heating from Storage and Auxiliary (Figure 10-16)	Closed	Open	Closed	Open	Off	On	On	Off	Closed
Pre-heating Water in Summer (Figure 10-17)	Open	Closed	Closed	Closed	On	Off	Off	On	Open

*Hot water also being heated if hot air temperature exceeds stored water temperature by preset amount, thereby actuating circulating pump.

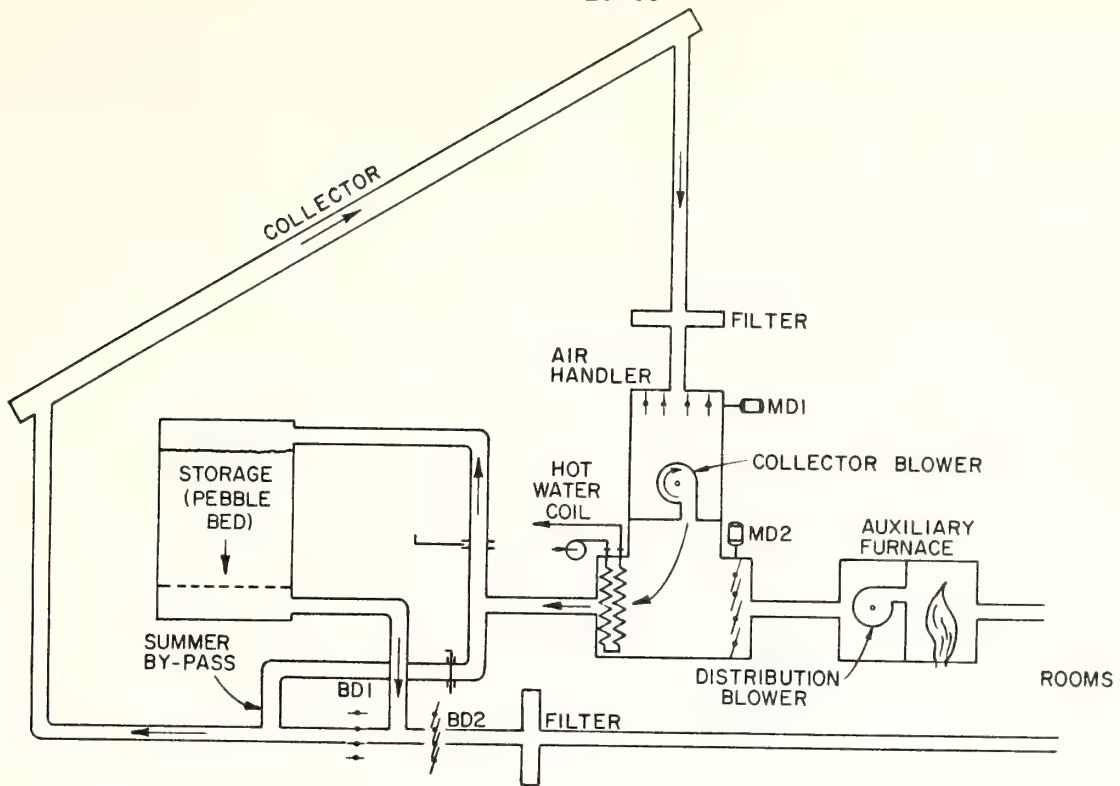


Figure 10-14. Storing Heat from Collectors

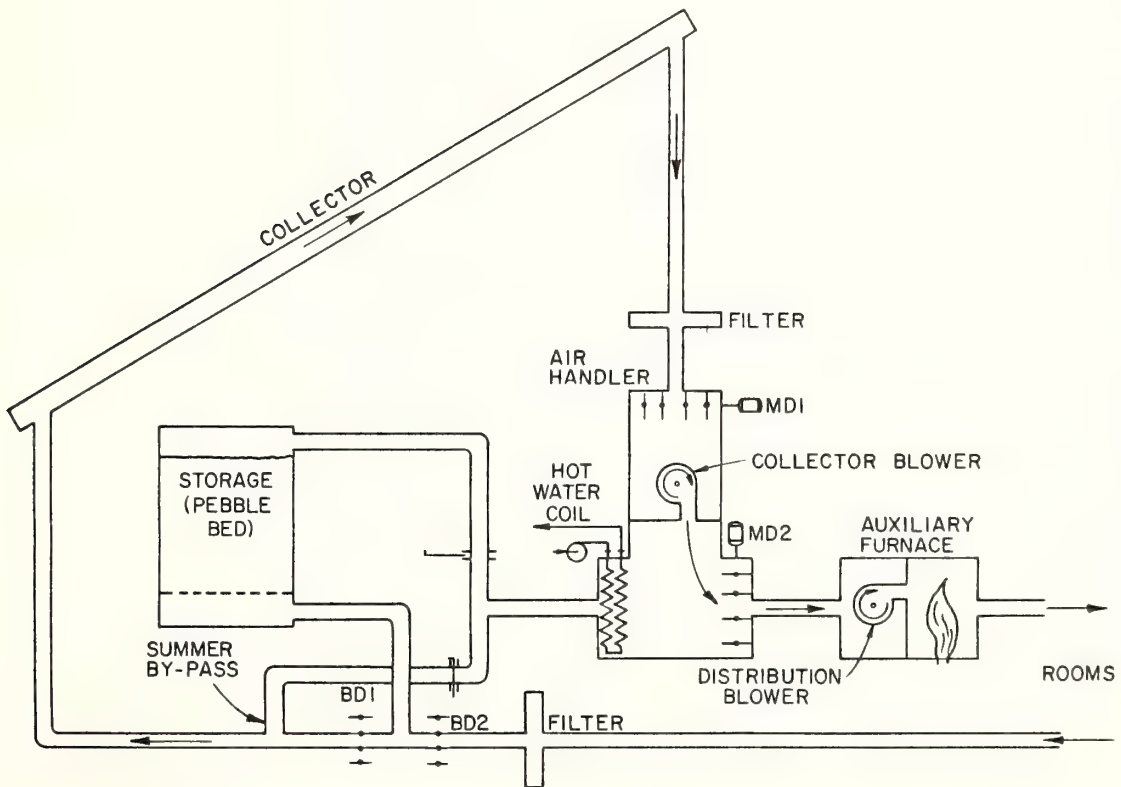


Figure 10-15. Heating Building from Collectors

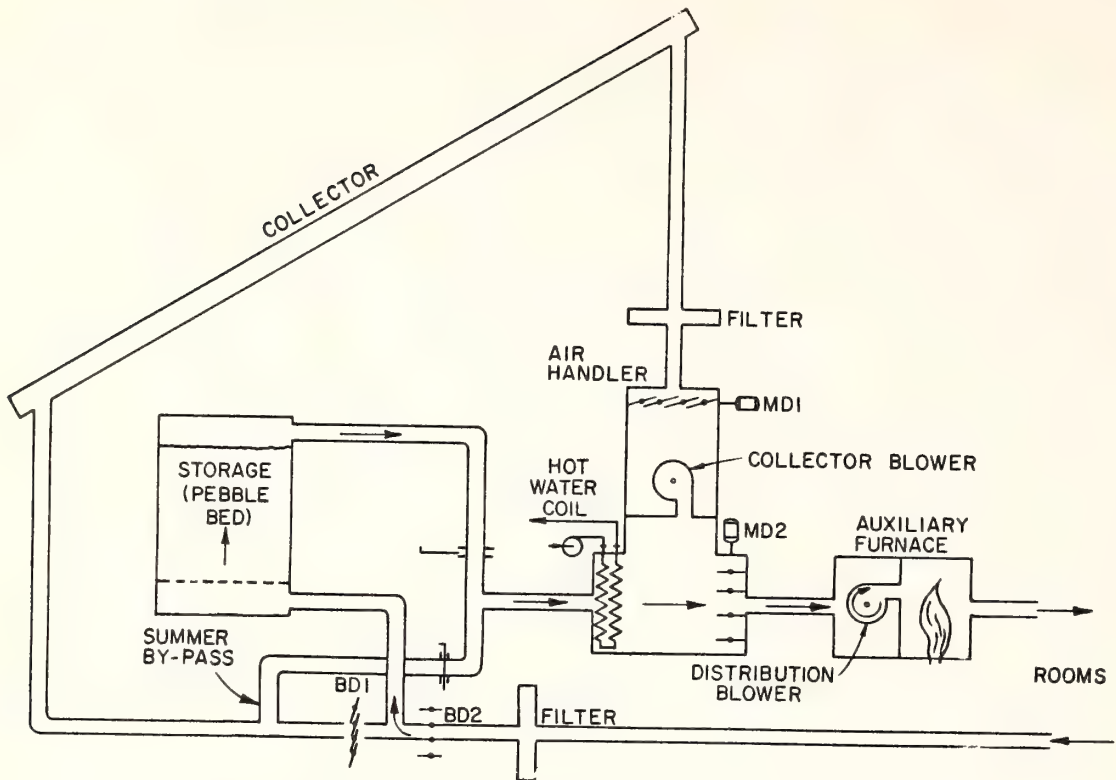


Figure 10-16. Heating Building from Storage Unit
(Also heating from auxiliary)

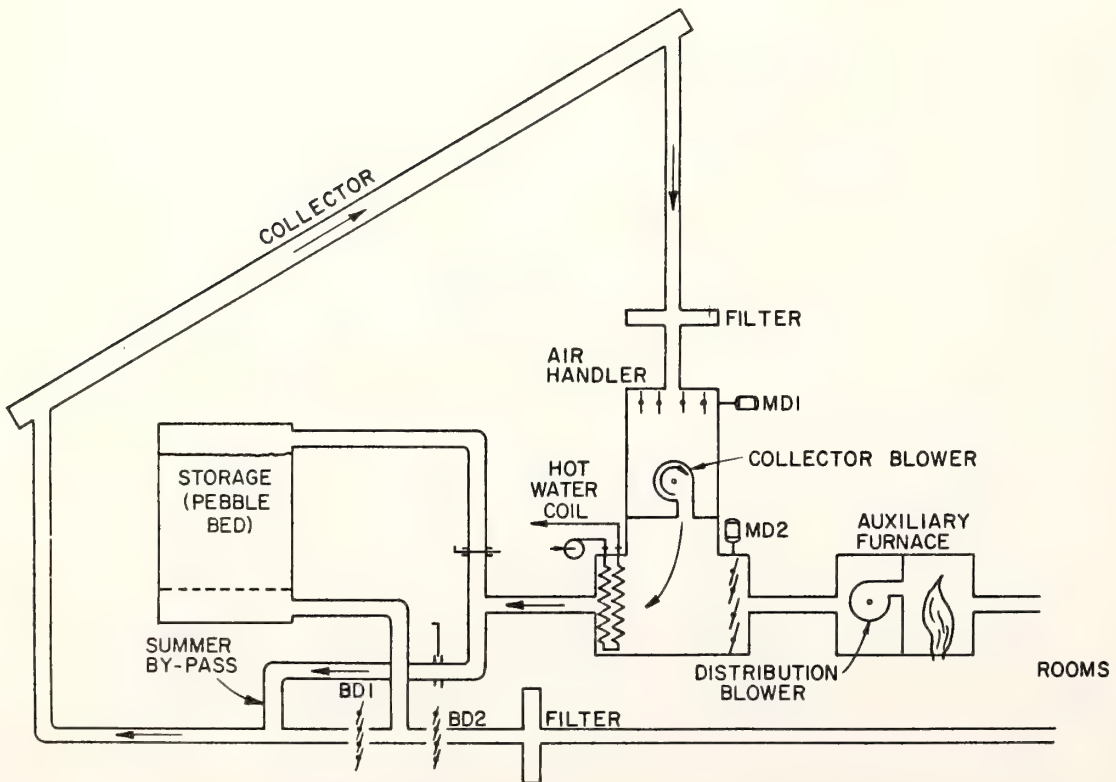


Figure 10-17. Service Hot Water Heating (Summer Operation)

the collector. When the hot sensor reaches a temperature about 45°F higher than the cold sensor, blower operation commences.* The signal from the controller that actuates the blower also positions dampers so that air passes from the collector to the hot end of the storage bed, through the pebbles, and from the cold end back to the collector. Figure 10-14 illustrates this mode of operation.

At a flow rate of about 2 cfm/ft² of collector, midday air temperatures from the collector usually range from 130°F to 170°F when air is being admitted to the collector at 70°F. As sundown approaches, the temperature declines, and when the preset turn-off difference is reached, usually about 20°F, the controller turns off the blower and repositions directional dampers.*

The pebble bed operates both as heat exchanger and heat-storage medium. The large heat exchange surface provided by the pebbles and the very low thermal conductivity from one pebble to another result in rapid transfer of heat from air to rock and a steep temperature gradient through the bed in the direction of air flow.

In most residential air-heating solar systems, solar heated water is also provided. A common finned coil is usually mounted at the air handler outlet leading to the pebble bed. During typical operation on sunny days, a small pump circulates water from the bottom of an insulated 50 to 100 gallon "pre-heat" tank, through the coil, and back to the top of the tank. If the temperature at collector outlet is higher than the tank bottom temperature by a preset amount, the pump is actuated.

*These control temperatures are the actual factory settings on a controller in a widely used air-heating solar system. As explained in Module 8, an "on" setting as low as 15° difference and an "off" setting of 2° to 3° will permit additional solar heat collection, but air delivery temperatures at these control points are lower than usually needed for space heating without auxiliary boosting.

The coil is sized so that a relatively small fraction of the collected solar heat is transferred to the water, usually enough to decrease the air temperature two or three degrees. The rise in water temperature per pass depends on the temperature in the tank as well as on the air temperature. Normally when heat is being stored in the pebble bed, water is also being heated.

Experience has shown that the coil location in Figures 10-13 to 10-17 is more satisfactory than in the duct between the collector and air handler, primarily because the possibility of freezing caused by nocturnal leakage of air through damper MD 1 is eliminated.

As shown in Figures 10-1, 10-2 and 10-3, solar heated water for domestic use is usually supplemented with auxiliary heat in a conventional water heater. The same arrangement is used with air-type collection systems. The two-tank design is nearly always employed if a fuel auxiliary is involved, but a single tank may be used if electric boosting is provided. In that case, solar heated water from the heat exchanger enters the tank at a point about one-third of the distance down from the top of the tank, and the electric element is positioned immediately above that point. Electrically heated water is therefore only in the upper third of the tank, and because of temperature stratification, the auxiliary heat does not adversely affect solar heat exchange. A tempering (mixing) valve should be installed in the service hot water supply line to prevent delivery of overheated water to the taps.

Daytime Space Heating

When heat is needed in the building at the same time solar energy is being collected, a room thermostat signals the control unit to move dampers and direct the flow of heated air from the collector directly to

the zones requiring heat, bypassing storage, as shown in Figure 10-15. In this mode, hot air passes from the collector through the collector blower, through the furnace with only its blower also in operation, and into the warm-air distribution system. Air circulates back from the rooms to the collector through conventional cold air return ducts. Either a motorized damper or a check damper (BD 2) (operated by slight pressure difference) is in this return duct. When room temperature requirements are satisfied, the thermostat breaks contact, and the storing mode is resumed.

When there is a high heat demand and when the temperature of the air being delivered from the collector is insufficient to meet the demand, room temperature will continue to decline. A lower thermostat set point then turns on the fuel in the auxiliary heater, thereby increasing the temperature of air supplied to the rooms. The full design capacity of the furnace will always provide sufficient heat to meet any demand, so the building temperature will be restored to the preset value.

Most commercially available warm-air furnaces for residential use contain a blower for circulation of warm air through the building via distribution ducts. In a typical all-air solar installation, the furnace blower is used in the normal manner for distributing warm air, supplied either from the collectors or from storage. The solar system blower operates when air is circulated through the collector either to storage or to distribution.

Space Heating from Storage

The third mode of operation, illustrated in Figure 10-16, is called for when heat is required in the building and solar collection is not taking place. Under these conditions, a room thermostat signals the

distribution blower in the furnace to operate and dampers to move so that room air will flow to the cold end of the storage bed, then from the hot end of the bed through the distribution blower and the furnace to the rooms via the air distribution system. Air leaving the hot end of the bed is only a few degrees below the rock temperature at that level. Heat is thus supplied to the room air by transfer from the heated pebbles.

If the pebble-bed discharge temperature is sufficient, air entering the rooms will provide enough heat to satisfy the thermostat, and after a sufficient period, the blower will cease operation. If, however, room temperature continues to drop, the auxiliary heat supply will be actuated by the lower thermostat set point, and auxiliary heat will also be supplied. This operation continues until room temperature rises to the upper thermostat set point and fuel and blower are shut off.

It can be seen from the above description that the use of solar heat is maximized by (1) collecting solar heat whenever moderate temperature delivery of 90°F to 100°F is possible; (2) utilizing even such low temperature heat, supplemented if necessary with auxiliary; (3) providing, by means of temperature stratification, high temperature storage even when the storage unit is only partially heated; (4) bypassing storage when heat is needed during sunny hours; (5) using auxiliary energy only as a supplement, not as a replacement for solar.

Summer Water Heating

So that the domestic hot water supply can be solar heated in the summer when no space heating is needed, the heat storage unit and heated space can be by-passed as shown in Figure 10-17. A manual or automatic damper is opened in a by-pass duct so that air is circulated in a closed

loop between collector, water heating coil, and the collector blower. Dampers MD 2, BD 1, and BD 2 in closed positions and a manual shut-off prevent flow of hot air to storage or to the rooms. The blower and water pump are actuated by the difference in temperature between collector outlet and hot water storage.

SINGLE-BLOWER DESIGN

Another damper arrangement does not require a furnace blower, so only the solar system blower is needed. Four motorized dampers are required (rather than two), but only two actuators are used. This system type is shown in Figure 10-18, with the blower and motorized dampers in an air handler cabinet. Although the cost of one blower and motor can be saved by this design, two additional dampers are required, the controls are more complicated, and airflow rates in the several modes are less adjustable. This arrangement is applicable when the air flow rate through the collectors is nearly equal to the air flow rate required in the heat delivery system to the rooms.

Additional details on the single-blower system and on the air handler utilized with it are presented in Module 8.

STORAGE SYSTEM

A pebble bed as a heat storage component has been discussed in Module 8. There are, however, some additional features that affect the use of a pebble bed in a complete solar heating system.

Heat flow in a pebble bed in the absence of air circulation is almost negligible, even in designs involving an air flow direction such that the hot end is at the bottom, the cold end at the top. Convective

heat transfer is minimized by the limited space for air movement between pebbles, and conduction is of little concern because of minimal pebble-to-pebble contact. Temperature profiles in a well-designed pebble bed do not change significantly overnight unless heat is being withdrawn by circulating air. At the bed center, for example, the temperature changes less than 1°F in 8 hours.

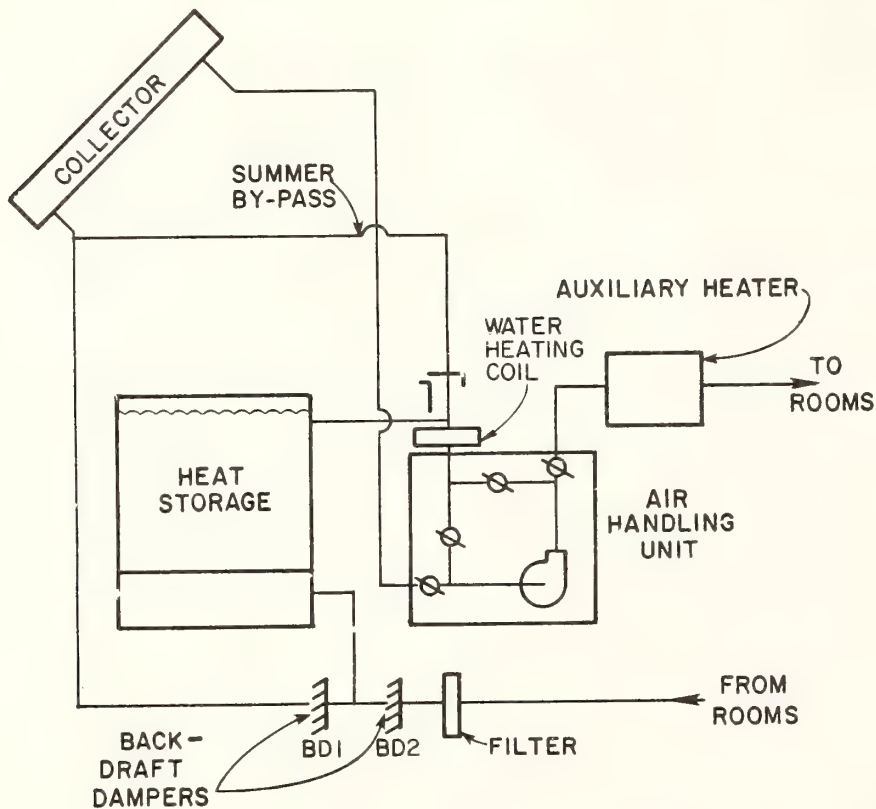


Figure 10-18. Single-Blower System

For similar reasons, heat loss from the pebble bed through the container walls can be easily controlled. Thick insulation is not required, both because of low conduction between the pebbles and the wall and also because the bed is usually in the building so that heat transferred through the container wall is not actually lost. Insulation with an R factor of 10 is more than satisfactory. Ordinary concrete block, reinforced concrete, and wood plank or plywood with gypsum board liner

may be used. A box formed of conventional insulated stud walls 3-1/2 in. thick with fiberglass between studs, and a similar top, is another option. In all of these arrangements, an overnight loss of heat through the walls from a completely charged storage unit should not exceed 1% of the thermal content.

The direction of air flow in the pebble bed is usually dictated by the position of heating system components rather than by thermal performance considerations. With storage depths of 6 ft or more, tests show that the performance of units having the hot end at the bottom is virtually the same as those heated from the top. The location of the blower, auxiliary furnace, and other system components, usually in the basement, may minimize duct lengths if the hot end is at the bottom of the pebble bed. But unless there is some practical reason to do otherwise, heated air should be supplied to the top of the bed so that there is minimum loss of temperature stratification and minimum heat loss from the bottom of the bed into the concrete base and underlying ground.

Horizontal-flow pebble beds have also been used, but the top cover of the bed must be in close contact with the pebbles so that air does not channel through an open space above the packing. There is some evidence that channeling of warm air through the pebbles in the upper part of the bed and of cool air through the lower part may occur. If a horizontal position cannot be avoided, vertical baffles or horizontal separators should be provided to prevent channeling, as shown in Module 8.

A pebble bed containing about one-half to three-fourths cubic foot of rock (weighing 50 to 75 pounds) per square foot of collector can store all of the heat deliverable from the collector on a completely sunny day. If all the heat collected during the day is delivered to

storage, maximum mid-winter collection of 800 Btu per square foot of collector would result in heating the pebbles from a starting temperature of 70 degrees to a final average temperature of 125°F to 150°F. During the winter months, in most practical systems, half to two-thirds of this heat would be placed in storage, the balance being used directly in the daytime. Essentially all of the stored heat would then usually be delivered to the heated space during the following night.

Studies have shown that the performance of a pebble-bed heat storage unit is dependent primarily on the mass of material and is comparatively insensitive to type of rock, dimensions of the bed, pebble size, and air flow rate. As a heat exchanger, effectiveness tends to increase with length of bed and with decreases in air flow rate and pebble size. But pressure loss and blower power requirements also increase with bed length and pebble fineness. Optimum design is thus an economic combination of these factors. Most commercially built pebble beds in single family houses are of nearly cubic shape, 5 to 7 feet dimension, containing 5 to 15 tons of locally sold screened gravel or crushed rock commonly used in concrete mixes. Rock size is usually 3/4 to 1-1/2 inch, with very little undersize material. Size uniformity is important for maintaining good heat transfer characteristics and low pressure loss and fan power consumption. Air velocity in typical pebble beds is usually not more than one-half foot per second, and pressure drop is generally less than 0.2 inch water gauge.

AIR FLOW RATES

An important design consideration is the flow rate through a solar air collector. Air delivery temperature decreases and solar collection efficiency increases with increased circulation rate. Fan power

requirement also rises with air flow, and there are practical limits to air circulation rates in the occupied space of a building. The efficiency of a solar air collector depends not only on volumetric flow, but also on air velocity. The type of manifolding, the length of travel of air in the collector, and the width of the air passages affect velocity.

In a typical commercial type of air collector, efficiency and pressure drop are at satisfactory levels with a flow of about 2 cfm/ft² of collector and a linear velocity of approximately 10 ft/sec. At an air flow rate of 2 cfm with a 13-ft air path through a solar collector having a 1/2-in. air passage, a pressure drop of approximately 0.25 in. of water is typical. Power requirements at this point of operation are moderate, less than 1 hp for circulating 1000 cfm of air through the collector and pebble bed. Total electric energy usage for solar energy collection, storage, and distribution in well-designed systems is less than ten percent of the solar energy supplied to use.

AUXILIARY HEAT

Auxiliary heat is usually supplied in solar air systems by use of a warm air furnace in series with the collector and storage units (Figures 10-15 and 10-16). This design permits maximum supply of solar energy by utilizing the solar system as a pre-heater of the air when heat requirements are greater than solar heat availability. Essentially all of the collected and stored solar heat can thus be used, even if at low temperature.

Warm air furnaces can be fueled with gas, oil, or propane, or they can be supplied with electricity. Control of auxiliary fuel or electricity is usually provided through a dual-stage house thermostat, the lower temperature contact actuating the fuel valve or electric switch.

Electric motors for blowers in warm air furnaces are frequently mounted in the cabinet through which air passes, and are thereby air-cooled. When used as an auxiliary heater in a solar air system, this motor usually operates in a warm air stream, occasionally at temperatures as high as 175° . A "type B" motor, suitable for operation at such temperatures, should therefore be used rather than the "type A" usually supplied in the furnace. The motor in the (solar) air handler, if also in the warm air stream, should be of the same type.

An air-to-air heat pump may also be used for supplying auxiliary heat, as shown in Figure 10-19. The condenser coil of the heat pump provides heat to the house air as the outdoor evaporator coil utilizes ambient air as the source of heat. Electric resistance back-up is also necessary for meeting high heating demands. Summer cooling can be supplied by reversing the evaporator and condenser functions so that electrically driven vapor-compression air-conditioning is provided in a conventional manner.

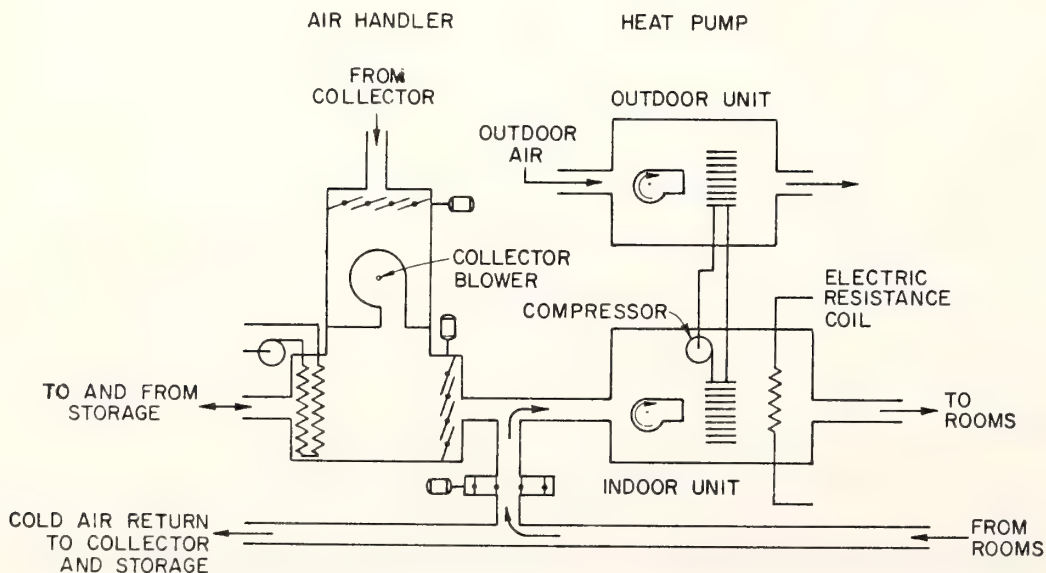


Figure 10-19. Solar Heating System with Air-to-Air Heat Pump Auxiliary

Because of pressure limitations in commercially available heat pumps, they are not usually operated as temperature boosters for solar heated air. An air temperature of 100°F may not be sufficient for heating the building, but supply of air at that temperature to the condenser coil would cause excessive pressure in the heat pump. For this reason, air is supplied to the heat pump from the house air return via the by-pass shown in Figure 10-19 rather than from the solar system. A motor-operated damper in the by-pass duct opens when this operating mode is required. This damper and the heat pump compressor and fans are controlled by the low temperature contact in the two-stage house thermostat, which simultaneously closes the damper at the air handler outlet. The electric resistance back-up coil is usually controlled by a temperature sensor in the heat pump system.

BLOWERS, DUCTS AND DAMPERS

An illustrative layout of a typical air-heating solar system (auxiliary heater and hot water tanks not shown) is presented in Figure 10-20. Although the positions of the several components will differ, the general arrangement in the two-blower system is usually as shown.

Important operating considerations in the air-type system are blower power requirements and air leakage. A well-designed air system has approximately equal pressure loss through the collectors and pebble bed, typically about 1/4-inch water gauge in each unit, although pressure differences across well-designed pebble beds are often as low as 0.1 inch W.G. With ducting, dampers, and filters, the total system pressure drop can approach one inch of water. This pressure difference is about twice that usually encountered in a conventional forced air

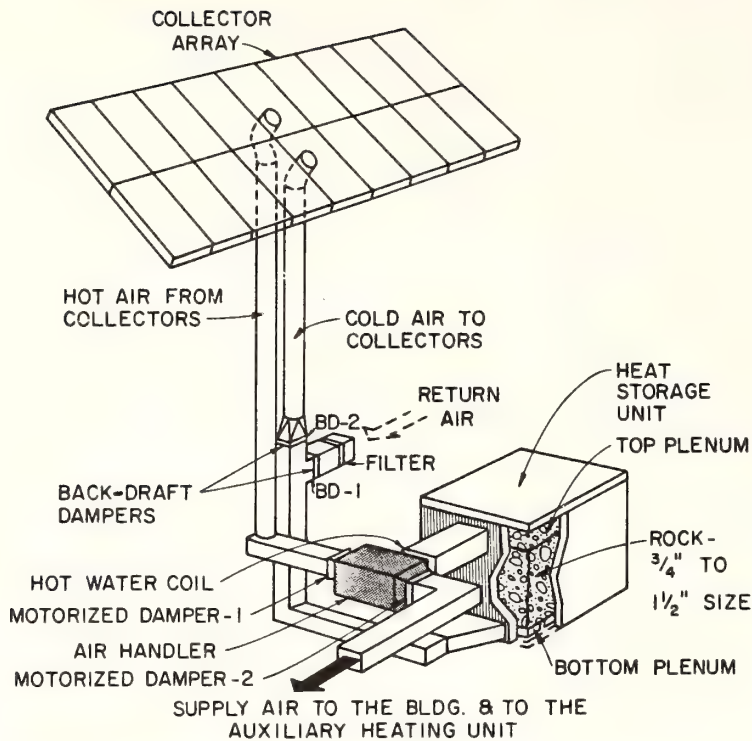


Figure 10-20. General Layout of Typical Air-Heating Solar System

distribution system, so additional blower power is required for its operation. A typical requirement in a conventional system is one-half to three-fourths horsepower for an air flow rate of 1000 to 1500 cfm. In a double-blower system, the collector blower motor is usually three-fourths to one horsepower, depending on collector area (300 to 700 square feet), and the distribution blower motor is of conventional size, usually about one-half horsepower. In a one-blower system, a one-horsepower motor will generally be required. The blowers also operate for longer periods than in the conventional system because of their use both for solar heat collection and for heat distribution. A one-inch water gauge pressure loss is about the maximum acceptable from the standpoint of blower power cost, although the electricity requirement is usually less than 10 percent of the solar heat supply.

Leakage of air in ducts, collectors, and storage is of greater concern in a solar heating system than in a conventional system because the pressure is higher, there is more ducting, the system operates for longer periods, and there may be more ducting through unheated space. Ducts should therefore be carefully inspected during installation and all joints should be sealed with a silicone sealing compound if sheet metal ducts are used. Ducts made of fiberglass board should be taped carefully at all corners and joints, and any perforations in the foil wrapper should be sealed. Insulation is needed to reduce heat loss through duct walls, particularly in unheated spaces such as attics. At least one inch of fiberglass with a rating of R-4 is recommended for duct insulation, with two inches for ducts in unheated spaces. Insulation may be either inside or outside sheet metal ducts.

It is especially important with a solar air system that a well-scheduled installation be made. More space and access must be provided in the building for ducting than for pipes in a liquid system. Ductwork and component assembly can be done at the same time that the distribution ducts and furnace are installed in a typical construction schedule. There must be provision for construction and installation space and for full access to the space for systems and components.

If fiberglass ductboard is used, it should not be in locations where it can be damaged by moving objects or occupants. Joints should be sealed with tapes or mastics recommended by the industry. Duct bends should be provided with turning vanes to reduce pressure loss. Ducts should be sized for air velocities between 600 and 800 feet per minute.

Blowers, dampers, and auxiliary heaters may be provided by a single solar system supplier or they may be purchased separately. Factory

mounting of motorized dampers and water heating coil in or on a blower cabinet, to provide complete hot air handling capability, substantially reduces site labor and increases quality and reliability of the installation. Blowers should be forward-curved squirrel cage type and preferably belt-driven to enable adjustments in air-flow rates. Direct-coupled blowers with motors in the air stream may be used and have the advantage of quieter operation, but disadvantages are the need for a type-B (high temperature) motor and the impossibility of blower speed adjustment. Flexible connections between blowers and ducts are recommended.

Louver-type dampers with neoprene or live silicone rubber seals are recommended for positive shutoff and smooth stroking. Damper drive motors should be located on the outside of ducts or air handlers and directly coupled to the damper shaft or through linkages. Special attention should be given to the linkages during installation to assure tight damper closure. In the one-blower system, damper pairs may be operated by the same drive motor so that one is closed when the other is open. Damper motors are available which operate on low voltage (24 volt) with spring returns, but greater reliability can be obtained by use of positive drive in two directions.

Back-draft dampers, used in ducts to prevent reverse air flow, may be of the flexible flap type or shutter type. They must be mounted to provide a positive seal against reverse airflow.

To prevent fouling and increased pressure loss in the pebble bed, filters should be installed in the air streams entering both ends of the storage unit. The filters should be changed or cleaned every few weeks during the first several months of operation to remove the initial dust from the system and building.

The complete solar heating installation requires carpenters or masonry workers to construct and fill the pebble-bed container, plumbers to connect the domestic water heating system, heating and sheet metal workers to install collectors, ducts, dampers, controls, and furnace, and electricians to wire blowers and dampers. Consequently, the general contractor and the solar system contractor should coordinate their activities so that each task is accomplished at the most appropriate and convenient stage during construction. Quality installation is an important requirement for obtaining good performance of an air-heating solar system.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 11

DESIGN PROCEDURES

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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OBJECTIVE

The objective of this module is to present a detailed method for estimating performance of solar heating systems. The trainee should be able to complete an f-chart system sizing procedure by "hand" calculation.

INTRODUCTION

Solar heating systems are sized to provide a desired fraction of the total annual heating load of the building. The desired fraction can be chosen arbitrarily or determined from economic analysis to minimize the annual cost of space heating and water heating with a solar-auxiliary system. Using approximate methods, the calculations lead to determination of collector area, and other components of the system are sized relative to the collectors. In detailed computations, collector area, storage size, heat exchanger size, and fluid flow rates through the collector are variable within practical ranges, and performance calculations include their variability.

Among several detailed methods devised for estimating system performance, the one described in this module is the "f-chart" method developed by Klein, Beckman and Duffie at the University of Wisconsin. The f-chart method is based on hour-by-hour simulations of performance of typical solar heating systems having wide ranges of system parameters in several geographic locations. Generalized correlation charts (f-charts) were developed for predicting the monthly fractions of the heating load supplied by solar energy. One chart was developed for

liquid-heating systems and another for air-heating systems. The f-chart method is applicable to system types described in Module 6.

Several assumptions were made in developing the f-charts:

1. Heat losses from storage are negligible.
2. Average domestic hot water demand is distributed throughout the hours of 6:00 a.m. to 1:00 a.m. in a nonuniform manner peaking at hours of 9:00 to 10:00 a.m. and 7:00 to 8:00 p.m. and the pattern is repeated every day.
3. Flat-plate collector performance can be characterized by two parameters, $F_R(\tau\alpha)_n$ and $F_R U_L$.

THE f-CHART METHOD

DESCRIPTION

There are separate f-charts for liquid- and air-type systems as shown on Figures 11-1 and 11-2, respectively. The coordinate axes X and Y characterize collector performance in relationship to the heating load for a specific month. In particular,

$$X = \frac{\text{collector losses}}{\text{monthly heating load}} = \frac{A_c F_R U_L (T_{\text{ref}} - T_a) \Delta t}{L} \quad (11-1)$$

and

$$Y = \frac{\text{collector heat gain}}{\text{monthly heating load}} = \frac{A_c F_R \overline{\tau\alpha} S}{L} \quad (11-2)$$

where X and Y are dimensionless, and the variables are explained below:

A_c is the total collector area, ft^2

$F_R U_L$ is the slope of the collector efficiency curve,
 $\text{Btu}/(\text{hr} \cdot \text{ft}^2 \cdot ^\circ\text{F})$

$F_R \overline{\tau\alpha}$ is the intercept on the collector efficiency curve,
corrected for incidence angle, dimensionless

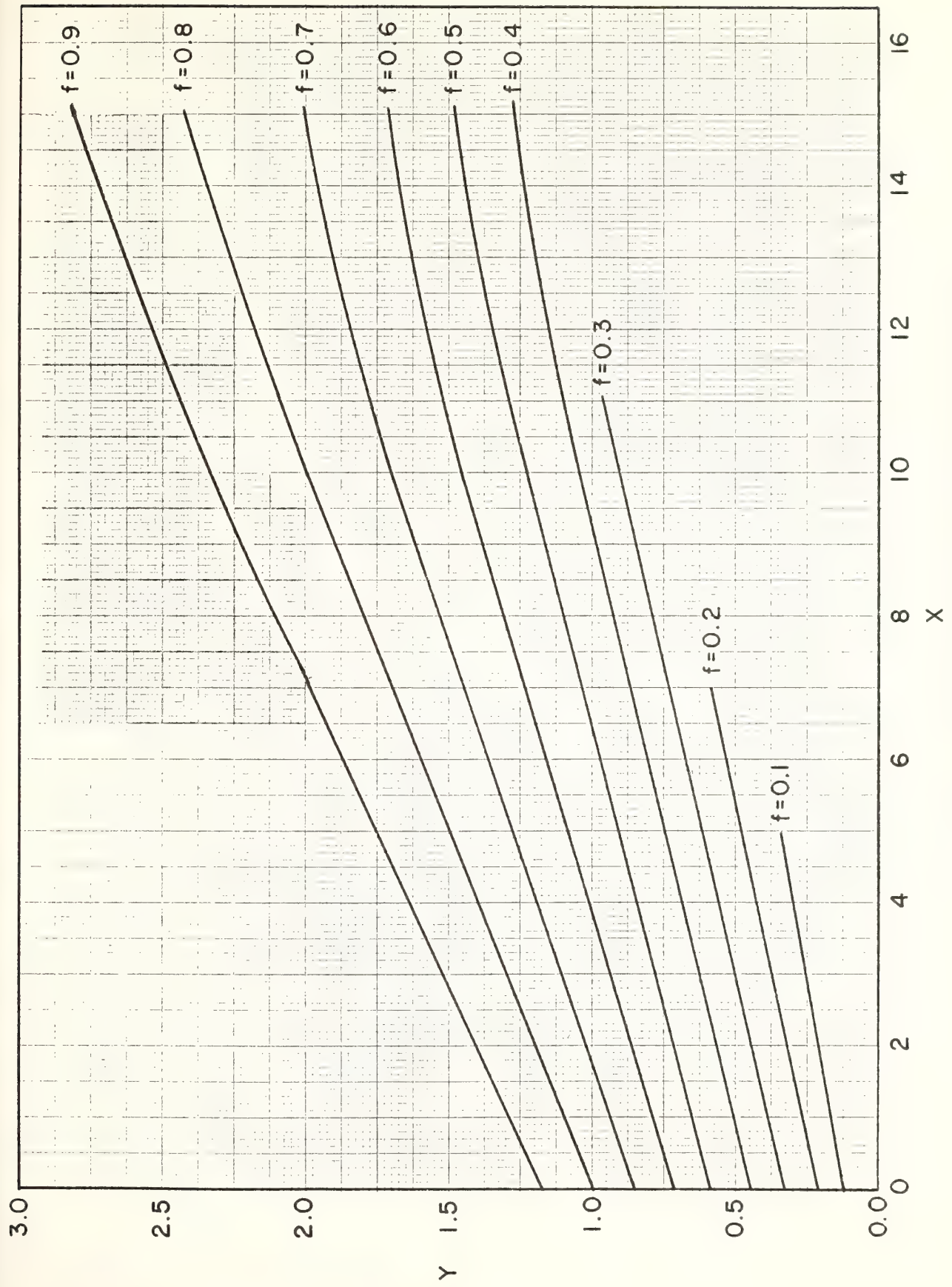
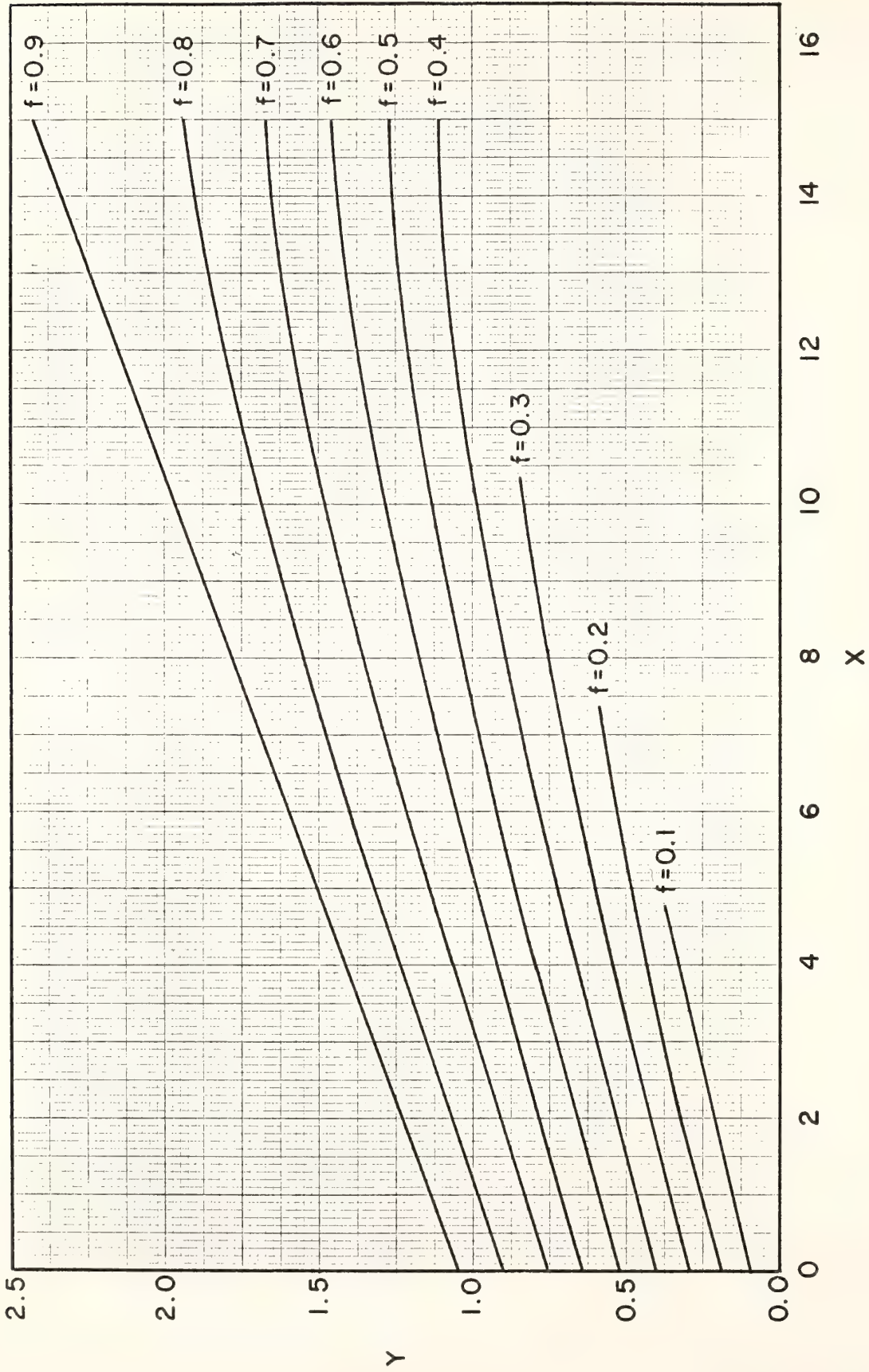


Figure 11-1. f-Chart for Liquid-Heating Solar Systems

Figure 11-2. f -Chart for Air-Heating Solar Systems

T_{ref}	is 212°F
T_a	is the average ambient (air) temperature for a specific month, °F
Δt	is the number of hours for a specific month, hr/mo
L	is the monthly space and water heating load, Btu/mo
S	is the average monthly solar radiation on a tilted collector per unit area, Btu/(ft ² ·mo)

The area is chosen arbitrarily, $F_R U_L$ and $F_R \overline{\alpha}$ are determined for a specific collector from performance curves, and T_a for specific cities is given in Module 3. The monthly heating load, L , is the sum of the DHW and space heating load, and S is determined from the monthly average daily solar radiation on a tilted surface (as described in Module 3), multiplied by the number of days in the month.

The f -charts of Figures 11-1 and 11-2 are solutions of bivariate equations. For liquid-based systems,

$$f = 1.029Y - 0.065X - 0.245Y^2 + 0.0018X^2 + 0.0215Y^3$$

for $0 \leq Y \leq 3$ and $0 \leq X \leq 18$. (11-3)

and for air-based systems,

$$f = 1.040Y - 0.065X - 0.159Y^2 + 0.00187X^2 - 0.0095Y^3$$

for $0 \leq Y \leq 3$ and $0 \leq X \leq 18$ (11-4)

CORRECTION FOR COLLECTOR/STORAGE HEAT EXCHANGER

A heat exchanger in the collector-to-storage flow circuit results in reduction of collector efficiency because the temperature of the collector return liquid is higher than it would otherwise be without a heat exchanger. The reduction of collector efficiency and net energy

collection is expressed as a fraction, F_R'/F_R , which is termed the heat exchanger factor.

Either the graph in Figure 11-3 or Equation (11-5) may be used to determine F_R'/F_R . The coordinates for Figure 11-3 are determined as follows:

$$x = \frac{C_c}{\epsilon_{cs} C_{min}}$$

in which

C_c is the collector fluid heat capacity rate, Btu/(hr·°F)

ϵ_{cs} is the heat exchanger effectiveness, dimensionless

C_{min} is the minimum of the collector or storage fluid capacitance ratio, Btu/(hr·°F)

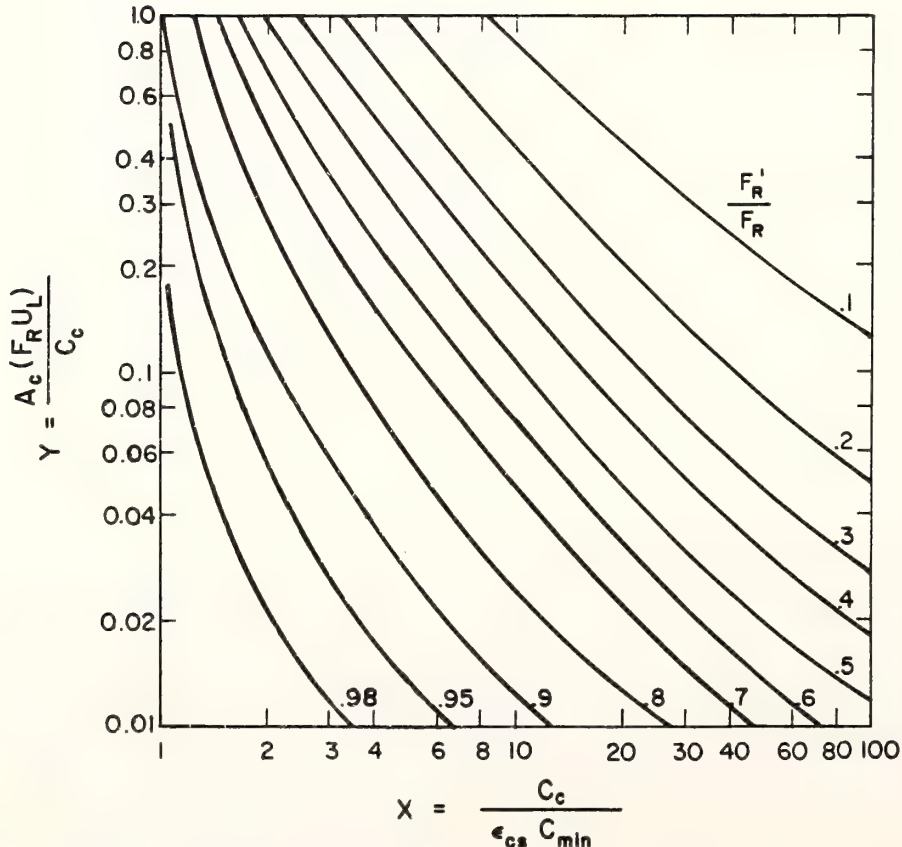


Figure 11-3. Heat Exchanger Factor F_R'/F_R

and,

$$y = \frac{A_c(F_R U_L)}{C_c}$$

where

A_c is the collector area, ft^2

$F_R U_L$ is the collector characteristic determined from the collector performance test data, as provided by the manufacturer, $\text{Btu}/(\text{hr} \cdot \text{ft}^2 \cdot ^\circ\text{F})$

The heat exchange factor may also be calculated from Equation (11-5)

$$\frac{F_R'}{F_R} = \frac{1}{1 + y(x-1)} \quad (11-5)$$

Corrections are made to X and Y values calculated in Equations (11-1) and (11-2) as follows.

$$X(\text{new value}) = X \cdot F_R'/F_R \quad (11-6)$$

$$Y(\text{new value}) = Y \cdot F_R'/F_R. \quad (11-7)$$

The new values of X and Y are used to determine monthly estimates of f from the chart, or Equations (11-3) and (11-4). In an air system, there is no heat exchanger between the collector and storage for space heating, so a correction factor is not applied.

CORRECTION FOR AIR FLOW RATE THROUGH COLLECTOR

Air collector efficiencies are sensitive to air flow rates through the collector, and correction factors are appropriate if the airflow rate is different from the collector manufacturer's recommendation. The correction factor is applied to the X value as follows:

$$X(\text{new value}) = X \cdot K_1 \quad (11-8)$$

where K_1 is determined from Figure 11-4 or from the Equation (11-9):

$$K_1 = \left(\frac{C_c}{2.13A_c} \right)^{0.28}$$

$$\text{for } 1 < C_c/A_c < 5 \quad (11-9)$$

where

C_c is capacitance flow rate through the collector

A_c is collector area

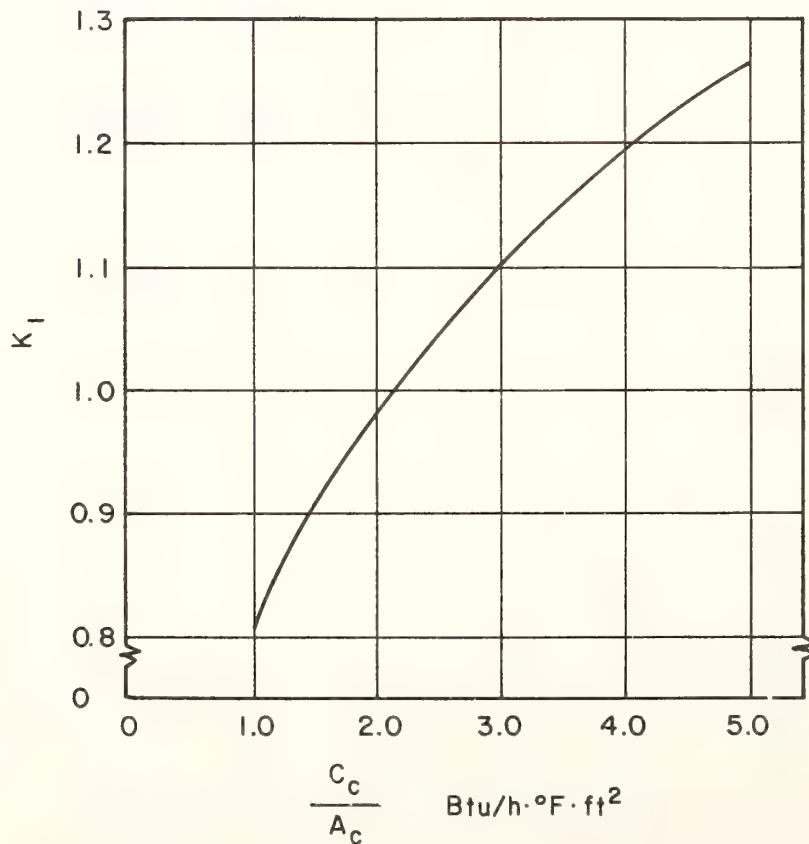


Figure 11-4. Collector Capacitance Factor (Air), K_1

CORRECTION FOR STORAGE

The normal storage capacity is assumed to be two gallons of water per square foot of collector [$16.7 \text{ Btu}/(\text{ft}^2 \cdot ^\circ\text{F})$] for a liquid system and 0.75 ft^3 of pebbles per square foot of collector [$15 \text{ Btu}/(\text{ft}^2 \cdot ^\circ\text{F})$] for an air system. When storage sizes differ from these values, the effect is determined by a correction to the X value in the following way:

$$X \text{ (corrected value)} = X \cdot K_2 \quad (11-10)$$

where K_2 is determined from Figure 11-5 or Equation (11-11) for water storage and Equation (11-12) for a pebble bed.

$$K_2(\text{water storage}) = \left(\frac{Mc_p}{16.7A_c} \right)^{-.25}$$

$$\text{for } 5 < Mc_p/A_c < 60 \quad (11-11)$$

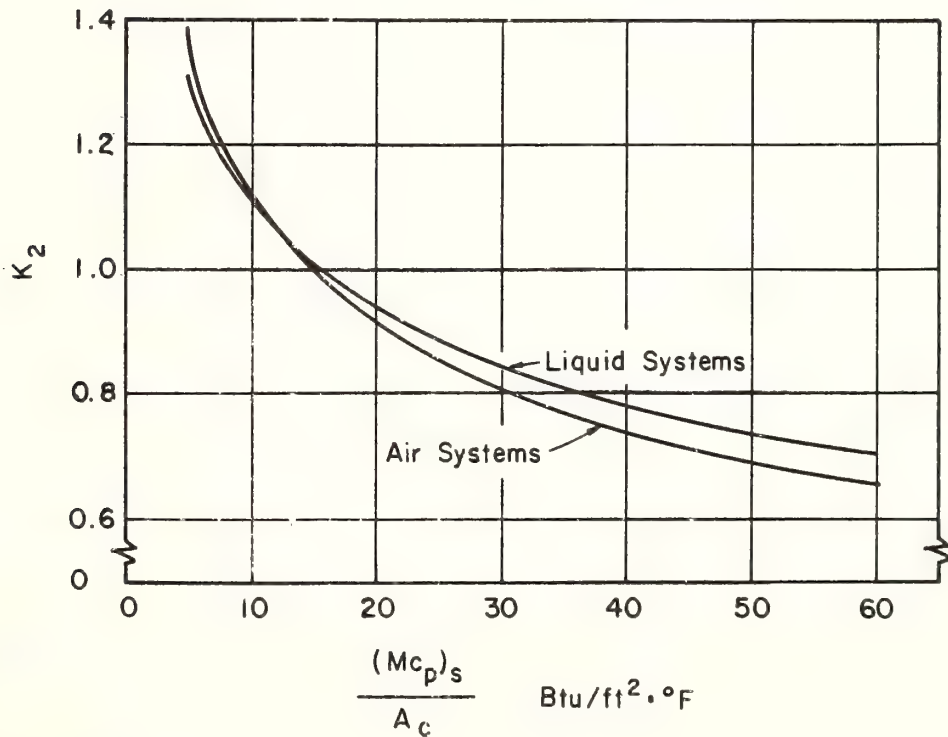


Figure 11-5. Storage Capacitance Factor, K_2

where

M is mass of storage (lb)

c_p is specific heat of storage mass (Btu/lb·°F)

A_c is collector area (ft²)

$$K_2(\text{pebble bed}) = \left(\frac{Mc_p}{15A_c} \right)^{-0.3}$$

$$\text{for } 5 < Mc_p/A_c < 60 \quad (11-12)$$

CORRECTION FOR LOAD HEAT EXCHANGER

One additional correction factor should be considered for a liquid-based system: the size of the load heat exchanger. A small load heat exchanger will affect the storage tank temperature and the temperature of the fluid circulated to the collector. The correction factor is expressed as a function of the load heat exchanger effectiveness in comparison to the UA of the building, and is applied to Y as follows:

$$Y (\text{corrected value}) = Y \cdot K_4 \quad (11-13)$$

where K_4 is determined from Figure 11-6 or Equation (11-14).

$$K_4 = 0.39 + 0.65 \exp \left(\frac{-0.139(UA)_{\text{bldg}}}{\varepsilon_L C_{\min}} \right)$$

$$\text{for } .5 < \frac{\varepsilon_L C_m}{UA} < 10 \quad (11-14)$$

where

ε_L is load heat exchanger effectiveness

C_{\min} is the smaller capacitance flow rate through the heat exchanger

$(UA)_{\text{bldg}}$ is the thermal conductance for the building

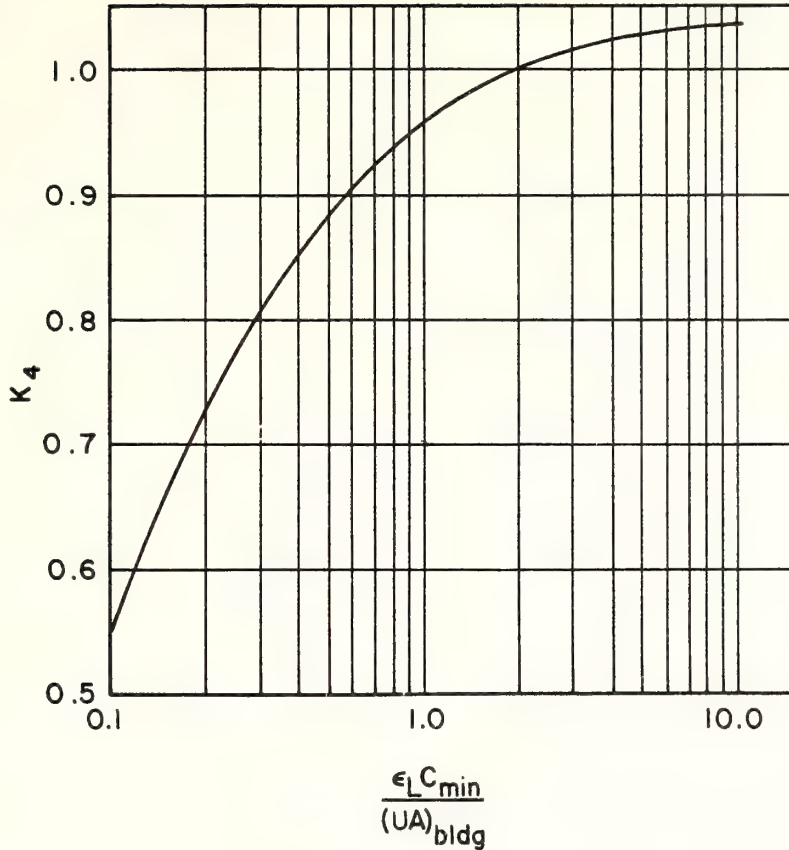


Figure 11-6. Load Heat Exchanger Factor, K_4

DOMESTIC WATER HEATING ONLY

The f-charts in Figures 11-1 and 11-2 were developed for combined space and water heating systems but may be used for solar domestic water heating systems by applying a correction to X . The correction factor depends upon temperatures of the hot water supply, cold water inlet, and ambient air and is applied in the following manner:

$$X \text{ (for DHW)} = X \cdot K_3 \quad (11-15)$$

where K_3 is determined from Figure 11-7 or Equation (11-16).

$$K_3 = \frac{1.18T_W + 3.86 T_M - 2.32 \bar{T}_a - 66.2}{202 - \bar{T}_a} \quad (11-16)$$

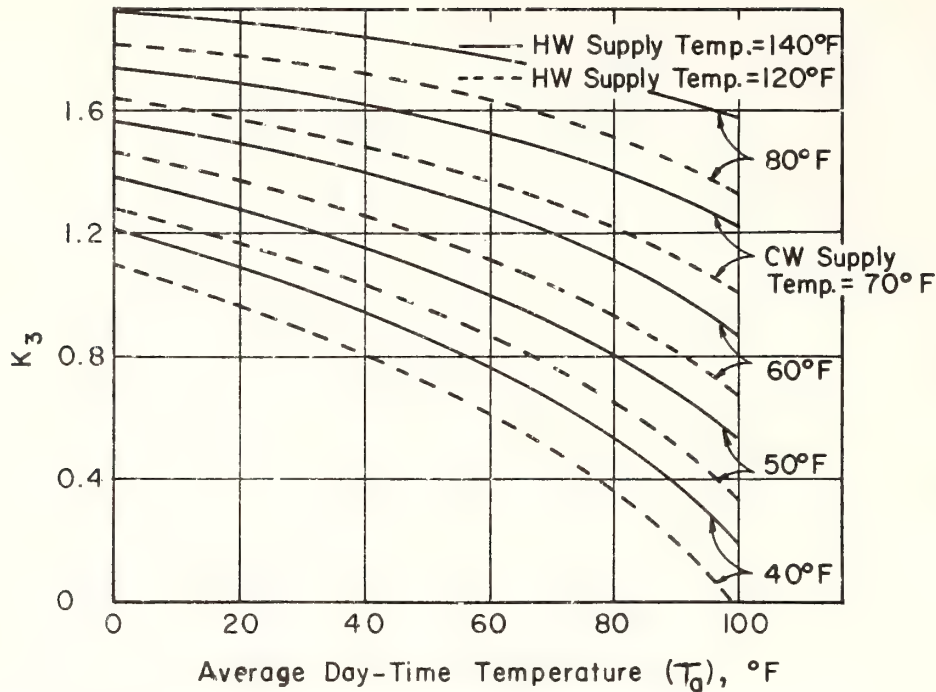


Figure 11-7. Hot Water Factor, K_3

where

T_W is the hot water supply temperature, °F
 T_M is the cold water inlet temperature, °F,
 (see Table 11-1 at end of this module)
 \bar{T}_a is the average ambient air temperature while
 collecting, °F.

CALCULATION PROCEDURE

The f-chart calculation procedure is outlined step-by-step and is followed by an example calculation. Space and DHW heating loads must be known or determined before beginning an f-chart performance analysis and it is helpful to use worksheets to organize the necessary computations. Separate worksheets are provided for air and liquid systems.

- Step 1. Solar System Data. Complete Worksheet A. Use available data from blueprints, specifications, inspections and handbooks. A heat load analysis for the building is required. For determining the UA of the building, refer to Worksheet B.
- Step 2. Monthly and Annual Heating/DHW Loads, L. Use Worksheet B. If the design heating load for the building is not available, an analysis is required. (Refer to Module 5).
- Step 3. Total Monthly Solar Radiation, S. Use Worksheet C.
- Step 4. Collector Performance Characteristics, $F'_R(\overline{\tau\alpha})$, $F'_R U_L$. Use Worksheet D. Lines 1, 2, 3 and 4 are transferred from Worksheet A.

Corrections to $F_R(\tau\alpha)_n$, $F_R U_L$ are necessary when the horizontal axis of the collector efficiency chart is based on fluid temperature other than the inlet temperature to the collector, T_i . Although collector test standards suggest use of T_i in expressing collector efficiencies, manufacturers do not always show collector characteristics in a uniform manner. The corrections to $F_R(\tau\alpha)_n$, and $F_R U_L$ for different cases are explained below.

Case 1. In $\frac{T^* - T_a}{T_i}$, T^* is T_{in} . (fluid inlet temperature)

No correction is needed

Case 2. If T^* is $\frac{T_i + T_{out}}{2}$, which is the average of the inlet and outlet temperatures,

$$F_R(\tau\alpha)_n \text{ (new value)} = F_R(\tau\alpha)_n \text{ (from efficiency curve)} \times \frac{1}{1 + \frac{F_R U_L A_c}{2C_c}}$$

where C_c is capacitance flow rate of the fluid through the collector, $(\dot{m} c_p)_c$, Btu/(hr·°F)

$$F_{R U_L} \text{ (new value)} = F_{R U_L} \text{ (from efficiency curve)} \times \frac{1}{1 + \frac{F_{R U_L} A_c}{2 C_c}}$$

$$C_c = \dot{m} c_p = (\text{volumetric flow rate})(\text{fluid density})(\text{heat capacitance})(\text{time conversion})$$

Case 3. If T^* is T_{out} (fluid outlet temperature) Btu/hr·°F

$$F_R(\tau\alpha)_n = F_R(\tau\alpha)_n \text{ (from efficiency curve)} \times \frac{1}{1 + \frac{F_{R U_L} A_c}{C_c}}$$

$$F_{R U_L} = F_{R U_L} \text{ (from efficiency curve)} \times \frac{1}{1 + \frac{F_{R U_L} A_c}{C_c}}$$

Incident Angle Modifier

Corrections to transmittance, τ , through the cover plates, and absorptance, α , for the absorber plate are necessary because of sun angle variations on the collector during the day. The $F_R(\tau\alpha)_n$ determined for normal incidence during collector testing must be corrected for an effective daily transmittance-absorptance product, $(\tau\alpha)$.

$$\frac{\overline{\tau\alpha}}{(\tau\alpha)_n} = \begin{cases} 0.91 & \text{for two cover plates} \\ 0.93 & \text{for one cover plate} \end{cases}$$

Collector/Storage Heat Exchanger Correction Factor

For air systems there is no correction. For liquid systems, correction is made to F_R in accordance with Module 7 (see Worksheet D Liquid Systems).

Step 5. Correction Factors K_1 , K_2 and K_4 . Use Worksheet E.

Step 6. System Performance parameters. Use Worksheet F.

Step 7. System Performance Calculations, f and F_{annual} . Use Worksheet G.

EXAMPLE 11-1

Estimate the performance of an air-type solar space and water heating system for a three bedroom house located in Denver, Colorado. A collector area of 300 ft^2 is planned. The house is wood framed with R-19 wall insulation and R-30 ceiling insulation. The overall dimensions of the house are 28 feet wide and 50 feet long and it is completely weatherized.

The heat loss calculations for the building have been made, and the overall UA (heat conductance) is $450 \text{ Btu}/(\text{hr} \cdot ^\circ\text{F})$. Using the time conversion from hour to day, the heating and domestic hot water load for the building is $10,800 \text{ Btu/DD}$. Complete Steps 1 through 7.

Step 1. Complete Worksheet A. (pp. 11-17, 11-18)

Step 2. Complete Worksheet B. (p. 11-19)

Step 3. Complete Worksheet C. (p. 11-20)

Step 4. Complete Worksheet D. (p. 11-21)

Step 6. Complete Worksheet F. (p. 11-23)

Step 7. Complete Worksheet G. (p. 11-24)

Answer. For collector area of 300 ft^2 , the air-type heating system will provide 76 percent of the total annual space and water heating load.

EXAMPLE 11-2

Estimate the performance of a liquid-type space and water heating system for a three bedroom house located in Fort Collins, Colorado. The UA for the building is $714 \text{ Btu}/(\text{hr} \cdot ^\circ\text{F})$ and hot water use is 80 gallons per day. The owner desires to install 500 ft^2 of RQP Company collectors at a tilt of 50° . Collector characteristics are:

$$F_R(\tau\alpha)_n = 0.73$$

$$F_R U_L = 0.54 \text{ Btu}/(\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F})$$

based on $(T_i - T_a)/I_T$.

A mixture of 30 percent ethylene glycol and water is specified for the collector loop. Storage volume is 1.5 gal/ft^2 of collector.

Answer. The solar system is estimated to provide 87.4 million Btu of useful heat for space and water heating, which is 68 percent of the total load.

Worksheet A
Sheet 1 of 2
AIR SYSTEMS

SOLAR SYSTEM DATA

Building Owner Mr + Mrs John Sunbody
 Address Denver, Colorado Ph. 482-0000
 Contractor Solar Construction Co Ph. 482-0001
 Type of System (liquid or air, H/DHW) Air, H/DHW

Site and Building Data

1. Location: Nearest City Denver Latitude 39.5° N
2. Building UA 450 Btu/(hr·°F)
3. DHW volume per day 80 gallons/day
4. Collector manufacturer XYZ Company
5. Collector area 300 ft²
6. Collector tilt 55 degrees
7. Tilt = latitude + 15 degrees
8. Collector orientation 0 degrees from south
9. Collector shading 0 % in December
10. Collector efficiency data
 - (a) $F_R(\tau\alpha)_n$ 0.69
 - (b) $F_R U_L$ 0.80 Btu/(hr·ft²·°F)
 - (c) Fluid temperature basis (circle one)

Case 1	T_i
Case 2	$\frac{T_i + T_{out}}{2}$
Case 3	T_{out}

11. Collector Fluid:

(a) Composition: Air(b) Flow rate G 600 ft³/min(c) Capacitance Flow rate $\frac{(\text{cfm}) \times (\text{density}) \times (\text{specific heat}) \times 60 \text{ m/hr}}{\text{collector area}}$ = 2.0 Btu/(hr·ft²·°F)

Storage Data

12. Storage medium Pebble Bed13. Unit volume 1.0 ft³/ft²14. Total volume (item 5 x 13) 300 ft³15. Storage Capacitance $\frac{(\text{vol}) \times (\text{density}) \times (\text{sp. heat})}{\text{collector area}}$ 20 Btu/(ft²·°F) =

Auxiliary Furnace/Boiler

Type Hot AirManufacturer LennoxRated Capacity 50,000 Btu/hrAuxiliary energy source Electricity

Auxiliary DHW Unit

Size 40 galAuxiliary energy source ElectricityHot water set temperature 140 °F

HEATING AND/OR DOMESTIC HOT WATER LOAD, L

Project Sanbody Residence

Denver

Month	1 Monthly Degree Days DD °F-days	2 Monthly Space Htg Load Q_s Btu/Mo.	No. of Days/ Mo. N	3 Vol. of DHW Used/Mo. Gal./Mo.	4 Temp. Water Main Sup. T_m °F	5 DHW Temp. Rise $T_{HW}-T_m$ °F	6 Monthly DHW Load Q_w Btu/Mo.	7 Total Heating Load L Btu/Mo.
Jan.	1132	$\times 10^6$ 12.2	31	2480	39	101	$\times 10^6$ 2.1	$\times 10^6$ 14.3
Feb.	938	10.1	28	2240	40	100	1.9	12.0
March	887	9.6	31	2480	43	97	2.0	11.6
April	558	6.0	30	2400	49	91	1.8	7.8
May	288	3.1	31	2480	55	85	1.8	4.9
June	66	0.7	30	2400	60	80	1.6	2.3
July	6	0.1	31	2480	63	77	1.6	1.7
Aug.	9	0.1	31	2480	64	76	1.6	1.7
Sept.	117	1.3	30	2400	63	77	1.5	2.8
Oct.	428	4.6	31	2480	56	84	1.7	6.3
Nov.	819	8.9	30	2400	45	95	1.9	10.8
Dec.	1035	11.2	31	2480	37	103	2.1	13.3
Totals	6283	67.9					21	89.5

$$Q_d = \underline{32,400} \text{ Btu/h}$$

(Given data or calculate as in Module 5)

$$\begin{aligned} \text{DTD} &= 70 - T_o \\ &= 70 - (-2) = 72^\circ \text{F} \end{aligned}$$

Where: T_o = 99% winter design temperature.
(From ASHRAE Fundamentals, or Table A5-10)

70°F = indoor design temperature

$$UA = \frac{Q_d}{\text{DTD}} = 450 \frac{\text{BTU}}{\text{hr} \cdot ^\circ \text{F}}$$

$$T_{HW} = \underline{140^\circ \text{F}}$$

1. From Table A5-10 or Figures A5-1 through A5-12

$$2. Q_s = (24)(UA)(\text{Degree Day})$$

$$3. (\text{Vol/day}) \times (\text{no. days/mo.}) = \underline{80} \text{ (gal./day)} \times (\text{no. days/mo.})$$

4. From Table 11-1 for selected cities

$$6. Q_w = (\text{vol. of water}) \times 8.34 \times 1 \times (T_{HW} - T_m).$$

$$7. L = Q_s + Q_w$$

TOTAL MONTHLY SOLAR RADIATION AVAILABLE, S

Project Sun body Residence
 Location Denver
 Collector Tilt Lat. +15°
 Nearest Data Site Denver

	1	2	3
Month	Monthly Avg. Daily Rad. on Tilt Surf. I_T Btu/(Day·ft ²)	No. of Days in month N	Tot. Monthly Radiation on Tilt Surf. S Btu/(Mo·ft ²)
Jan.	1975	31	61,225
Feb.	2051	28	57,596
March	2008	31	62,248
April	1816	30	54,480
May	1673	31	51,863
June	1710	30	51,300
July	1679	31	52,049
Aug.	1876	31	58,156
Sept.	2046	30	61,380
Oct.	2064	31	63,984
Nov.	1864	30	55,920
Dec.	1768	31	54,808

COLLECTOR COMBINED PERFORMANCE CHARACTERISTICS, $F_R(\overline{\tau\alpha})$, $F_R U_L$ PROJECT Sunbody Residence

Collector Efficiency Data from Worksheet A (lines 10(a), (b))

1. Intercept, $F_R(\tau\alpha)_n = \underline{0.69}$

2. Slope, $F_R U_L = \underline{0.80}$

Reference Temperature Basis: 1. t_{in} , 2. $\frac{t_{in} + t_{out}}{2}$, 3. t_{out}

3. Collector area, $A_c = \underline{300} \text{ ft}^2$

4. Collector volumetric flow rate (Worksheet A, 11(d))

$\underline{600} \text{ ft}^3/\text{min}$

Correction to t_{in} basis

5. Case 1: (no correction) $F_R(\tau\alpha)_n = \underline{0.69}$

$F_R U_L = \underline{0.80}$

6. Case 2: $F_R(\tau\alpha)_n = F_R \tau\alpha \times \left[\frac{1}{1 + \frac{F_R U_L A_c}{2 C_c}} \right] = \underline{N/A}$

$F_R U_L = F_R U_L \times \left[\frac{1}{1 + \frac{F_R U_L A_c}{2 C_c}} \right] = \underline{N/A}$

$$C_c = \dot{m}_{c_p} = (\text{volumetric flow rate})(\text{density})(\text{time conversion})(\text{specific heat})$$

where: for liquids, density = (8.34 lb/gal) x

(specific gravity) for air, density = 0.075 lb/ft³

at 70° and 1 atm. specific heat = 0.24 Btu/lb·°F

7. Case 3: $F_R(\tau\alpha)_n = F_R(\tau\alpha)_n \times \left[\frac{1}{1 + \frac{F_R U_L A_c}{C_c}} \right] = \underline{N/A}$

$F_R U_L = F_R U_L \times \left[\frac{1}{1 + \frac{F_R U_L A_c}{C_c}} \right] = \underline{N/A}$

8. Incident Angle Modifier, $\frac{(\overline{\tau\alpha})}{(\tau\alpha)_n} = \begin{cases} .91, & \text{for two cover plates} \\ .93, & \text{for one cover plate} \end{cases}$

9. $F_R(\overline{\tau\alpha}) = F_R(\tau\alpha)_n \times \frac{\overline{\tau\alpha}}{(\tau\alpha)_n} = \underline{0.69} \times \underline{0.93} = \underline{0.64}$

Worksheet E
AIR SYSTEMSCORRECTION FACTORS, K_1 , K_2 PROJECT Sunbody ResidenceCollector Flow Factor, K_1

1. Air Flow rate (Worksheet A, line 11(b)) = 600 cfm
2. Capacitance Flow rate (from Worksheet A, line 11(c)) 20 Btu/(h·ft²·°F)
3. K_1 = (from Figure 11-4) = 0.98

Storage Mass Capacitance Factor, K_2

4. Storage Capacitance (Worksheet A, line 15 =) 20 Btu/ft²·°F
5. K_2 = (from Figure 11-5) = 0.92

SYSTEM PERFORMANCE PARAMETERS X, Y

PROJECT Sunbody House
LOCATION Denver Lat = 39.4°

Month	1 Tot. Monthly Radiation on Tilt Surf. S Btu/(Mo·ft ²)	2 Total Heating Load L Btu/Mo. x 10 ⁶	3 S/L	4 Y F ₁ ·[3]	5 Mo. Av. Temp. T _a °F	6 212-T _a °F	7 Tot. Hrs in Mo. Δ time hr.	8 X
Jan.	61,225	14.3	.00428	0.82	182	182	744	2.05
Feb.	51,596	12.0	.00480	0.92	180	180	672	2.18
March	62,248	11.6	.00537	1.03	175	175	744	2.43
April	54,480	7.8	.00698	1.34	164	164	720	3.28
May	51,863	4.9	.01058	2.03	155	155	744	5.09
June	51,300	2.3	.02230	4.28	146	146	720	9.89
July	52,049	1.7	.03062	5.88	139	139	744	13.16
Aug.	58,156	1.7	.03421	6.57	140	140	744	13.26
Sept.	61,380	2.8	.02142	4.21	149	149	720	8.29
Oct.	63,984	6.3	.01016	1.95	160	160	744	4.09
Nov.	55,920	10.8	.00518	0.99	173	173	720	2.50
Dec.	54,808	13.3	.00412	0.79	180	180	744	2.18

1. From Worksheet C, Col. 3

2. From Worksheet B, Col. 7

$$4. Y = \frac{A_c F_R (\overline{\tau\alpha}) S}{L} = F_1 \cdot \frac{S}{L}$$

5. From Module 3

$$8. X = \frac{A_c F_R U_L (T_{ref} - T_a) \Delta time}{L} \times K_1 \times K_2$$

$$= F_2 \cdot [(6) \cdot (7)] \cdot [2]$$

$$\begin{aligned} A_c &= 300 \text{ ft}^2 \\ F_R \overline{\tau\alpha} &= 0.64 \text{ (Wksht D)} \\ F_R U_L &= 0.80 \text{ (Wksht D)} \\ K_1 &= 0.98 \text{ (Wksht E)} \\ K_2 &= 0.92 \text{ (Wksht E)} \\ F_1 &= A_c F_R \overline{\tau\alpha} = 192 \\ F_2 &= A_c F_R U_L K_1 K_2 = 216.4 \end{aligned}$$

FRACTION OF TOTAL HEATING LOAD SUPPLIED BY SOLAR ENERGY, F_{Annual}

PROJECT Sunbody Residence

COLLECTOR AREA 300 ft²
LOCATION Denver

	1	2	3	4	5
Month	Tot. Mo. Htg. Load $L \times 10^6$ Btu/mo.	System Parameter X	System Parameter Y	Solar Fraction/ mo. f	Actual Solar en/mo $E \times 10^6$ Btu/mo.
Jan.	14.3	2.05	0.82	0.62	8.9
Feb.	12.0	2.18	0.92	0.68	8.6
March	11.6	2.43	1.03	0.75	8.7
April	7.8	3.28	1.34	0.89	6.9
May	4.9	5.09	2.03	1.00	4.9
June	2.3	9.89	4.28	1.00	2.3
July	1.7	13.16	5.88	1.00	1.7
Aug.	1.7	13.26	6.52	1.00	1.7
Sept.	2.8	8.29	4.21	1.00	2.8
Oct.	6.3	4.09	1.95	1.00	6.3
Nov.	10.8	2.50	0.99	0.71	7.7
Dec.	13.3	2.18	0.79	0.58	7.7

$$L_{\text{tot}} = 89.5 \times 10^6 \text{ Btu}$$
$$E_{\text{tot}} = 67.8 \times 10^6 \text{ Btu}$$
$$F_{\text{Annual}} = \frac{E_{\text{tot}}}{L_{\text{tot}}} = \frac{67.8}{89.5} = 0.76$$

1. From Worksheet B
2. From Worksheet F, Column 8
3. From Worksheet F, Column 4
4. From "f chart", Figure 11-2 or Equation (11-4)

SOLAR SYSTEM DATA

Building Owner Mr + Mrs John Sunbody
 Address 736 Sunshine Ave Fort Collins Colo Ph. 482-0000
 Contractor Solar Construction Co Ph. 482-0001
 Type of System (liquid or air, H/DHW) Liquid, M/DHW

Site and Building Data

1. Location: Nearest City Fort Collins Latitude 40.6°N
2. Building UA 714 Btu/(hr·°F)
3. DHW volume per day 80 gallons/day
4. Collector manufacturer RAP Company
5. Collector area 500 ft²
6. Collector tilt 50 degrees
7. Tilt = latitude + 10 degrees
8. Collector orientation 0 degrees from south
9. Collector shading 0 % in December
10. Collector efficiency data
 - (a) $F_R(\tau\alpha)_n$.13
 - (b) $F_R U_L$.54 Btu/(hr·ft²·°F)
 - (c) Fluid temperature basis (circle one)
 - Case 1 (T_i)
 - Case 2 $\frac{T_i + T_{out}}{2}$
 - Case 3 T_{out}
11. Collector Fluid:
 - (a) Composition: water ethylene glycol, 30%
 - (b) Specific heat, c_p (from Mod. 4) 0.90 Btu/(lb·°F)
 - (c) Fluid density, ρ (from Mod. 4) 8.92 lb/gal
 - (d) Flow rate G 10 gal/min

Storage Data

12. Storage medium water
13. Unit volume 1.5 gal/ft² or ft³/ft²
14. Total volume (item 5 x 13) 750 gal or ft³

15. Specific heat of storage material c_p 1 Btu/(lb·°F)
 16. Density 8.34 lb/gal
 17. Total mass (14 x 16) M 6255 lb.
 18. Total heat capacity (17 x 15)
 $C_s = M c_p$ 6255 Btu/°F
 19. Total heat capacity per unit
 collector area (18 ÷ 5) 12.5 Btu/(ft²·°F)

Heat Exchangers

20. Collector/storage type Counter flow and
 manufacturer Young Radiator
 21. Storage loop flow rate 15 gal/min
 22. Heat exchange effectiveness ϵ_{cs} 0.75
 23. Load heat exchanger type Cross-flow, water-air and
 manufacturer Young Radiator
 24. Load loop flow rate 12 gal/min
 25. Building air supply flow rate 1200 ft³/min
 26. Heat exchanger effectiveness ϵ_L 0.75

DHW Pre-heater

27. Collector/storage heat exchanger type Counter flow and
 manufacturer Young Radiator
 28. Collector loop flow rate - gal/min
 29. Heat exchanger effectiveness ϵ_{Hw} 0.7
 30. Storage volume 80 gal
 31. Storage mass, $M_{St.}$ (line 30 x 8.34) 667 lb

Auxiliary Furnace/Boiler

32. Type Cold Boiler
 33. Manufacturer RQP Company
 34. Rated capacity 100,000 Btu/hr
 35. Auxiliary energy source electricity

Auxiliary DHW Unit

36. Size 40 gal
 37. Auxiliary energy source electric
 38. Hot water set temperature 140 °F

HEATING AND/OR DOMESTIC HOT WATER LOAD, L

Project Sunbody

Residence

$$Q_d = \frac{51,410}{\text{Btu/h}}$$

(Given data or calculate as in Module 5)

$$\begin{aligned} \text{DTD} &= 70 - T_o \\ &= 70 - (-2) = 72^\circ\text{F} \end{aligned}$$

Where: T_o = 99% winter design temperature.
(From ASHRAE Fundamentals, or Table A5-10)

70°F = indoor design temperature

$$UA = \frac{Q_d}{\text{DTD}} = \frac{714 \text{ Btu/(hr} \cdot ^\circ\text{F)}}{140^\circ\text{F}}$$

$$T_{\text{HW}} = 140^\circ\text{F}$$

Month	1 Monthly Degree Days DD °F-days	2 Monthly Space Htg Load Q_s Btu/Mo.	3 Vol. of DHW Used/Mo. Gal./Mo.	4 Temp. Water Main Sup. T_m °F	5 DHW Temp. Rise $T_{\text{HW}} - T_m$ °F	6 Monthly DHW Load Q_w Btu/Mo.	7 Total Heating Load L Btu/Mo.
Jan.	1138	$\times 10^6$ 19.4	2480	39	101	$\times 10^6$ 2.1	21.5
Feb.	938	16.1	2240	40	100	1.9	18.0
March	887	15.2	2480	43	97	2.0	17.2
April	558	9.6	2400	49	91	1.8	11.4
May	288	5.0	2480	55	85	1.8	6.8
June	66	1.1	2400	60	80	1.6	2.7
July	6	.1	2480	63	77	1.6	1.7
Aug.	9	.2	2480	64	76	1.6	1.8
Sept.	117	2.0	2400	63	77	1.5	3.5
Oct.	428	7.3	2480	56	84	1.7	9.0
Nov.	819	14.0	2400	45	95	1.9	15.9
Dec.	1035	17.7	2480	37	103	2.1	14.8
					Total	21.0	129.3

- From Table A5-10 or Figures A5-1 through A5-12
- $Q_s = (24)(UA)(\text{Degree Day})$
- $(\text{Vol/day}) \times (\text{no. days/mo.}) = \frac{80}{\text{gal./day}} \times (\text{no. days/mo.})$
- From Table 11-1 for selected cities.
- $Q_w = (\text{vol. of water}) \times 8.34 \times 1 \times (T_{\text{HW}} - T_m)$.
- $L = Q_s + Q_w$

TOTAL MONTHLY SOLAR RADIATION AVAILABLE, S

Project Sun body Residence
 Location Fort Collins
 Collector Tilt Lat + 15°
 Nearest Data Site Boulder

	1	2	3
Month	Monthly Avg. Daily Rad. on Tilt Surf. I_T Btu/(Day·ft ²)	No. of Days in month N	Tot. Monthly Radiation on Tilt Surf. S Btu/(Mo·ft ²)
Jan.	1439	31	44,609
Feb.	1533	28	42,924
March	1909	31	59,179
April	1663	30	49,890
May	1425	31	44,175
June	1491	30	44,730
July	1535	31	47,585
Aug.	1458	31	45,198
Sept.	1702	30	51,060
Oct.	1659	31	51,429
Nov.	1507	30	45,210
Dec.	1418	31	43,958

COLLECTOR COMBINED PERFORMANCE CHARACTERISTICS, $F_R'(\overline{\tau\alpha}) \cdot F_R' U_L$ PROJECT Sun body Residence

Collector Efficiency Data from Worksheet A (lines 10(a), (b))

1. Intercept, $F_R(\tau\alpha)_n = \underline{0.73}$

2. Slope, $F_R U_L = \underline{0.54}$

Reference Temperature Basis: 1. t_{in} , 2. $\frac{t_{in} + t_{out}}{2}$, 3. t_{out}

3. Collector area, $A_c = \underline{500} \text{ ft}^2$

4. Collector volumetric flow rate (Worksheet A, 11(d))

10 gal/min or ft³/min

Correction to t_{in} basis

5. Case 1: (no correction) $F_R(\tau\alpha)_n = \underline{0.73}$

$F_R U_L = \underline{0.54}$

6. Case 2: $F_R(\tau\alpha)_n = F_R \tau\alpha \times \left[\frac{1}{1 + \frac{F_R U_L A_c}{2 C_c}} \right] = \underline{N/A}$

$F_R U_L = F_R U_L \times \left[\frac{1}{1 + \frac{F_R U_L A_c}{2 C_c}} \right] = \underline{N/A}$

$$C_c = \dot{m} c_p = (\text{volumetric})(\text{density})(\text{time conversion})(\text{specific heat})$$
$$= 10 \times 8.92 \times 60 \times 0.90 = \underline{4817 \text{ Btu/(hr} \cdot ^\circ\text{F)}}$$

where: for liquids, density = (8.34 lb/gal) x
(specific gravity) for air, density = 0.075 lb/ft³
at 70° and 1 atm. specific heat = 0.24 Btu/lb·°F

7. Case 3: $F_R(\tau\alpha)_n = F_R(\tau\alpha)_n \times \left[\frac{1}{1 + \frac{F_R U_L A_c}{C_c}} \right] = \underline{N/A}$

$F_R U_L = F_R U_L \times \left[\frac{1}{1 + \frac{F_R U_L A_c}{C_c}} \right] = \underline{N/A}$

8. Incident Angle Modifier, $\frac{(\tau\alpha)}{(\tau\alpha)_n} = \begin{cases} .91 & \text{for two cover plates} \\ .93 & \text{for one cover plate} \end{cases}$

Collector Loop Heat Exchanger Modifier, $\frac{F'_R}{F_R}$

9. For air systems and liquid systems without a collector/storage heat exchanger, $\frac{F'_R}{F_R} = 1$

Capacitance Rate:

10. $C_c =$ (from line 6) $=$ 4817 Btu/(hr·°F)
11. $C_s =$ (calc. as for C_c above) $=$ 1506 Btu/(hr·°F)
12. $C_{\min} =$ (lesser of C_c and C_s) $=$ 4817 Btu/(hr·°F)
13. Collector Storage Heat Exchanger Effectiveness, $\epsilon_{cs} =$ 0.75
14. $x = \frac{C_c}{\epsilon_{cs} C_{\min}} = \frac{4817 / 0.75 (4817)}{\text{(from Worksheet A, line 22)}} =$ 1.33
15. $y = \frac{A_c (F_R U_L)}{C_c} = \frac{500 (.54)}{4817} =$.056
16. $\frac{F'_R}{F_R} = \frac{1}{1 + y(x-1)} =$.98
17. $F'_R(\overline{\tau\alpha}) = F_R(\tau\alpha)_n \times \frac{(\overline{\tau\alpha})}{(\tau\alpha)_n} \times \frac{F'_R}{F_R} =$ (0.73)(.93)(.98) = .67
18. $F'_{RU_L} = F_{RU_L} \times \frac{F'_R}{F_R} =$ (.54)(.98) = .53

CORRECTION FACTORS, K_2 , K_4 PROJECT Sunbody ResidenceStorage Mass Capacitance Factor, K_2

Note: M includes hot water storage volume where it is solar heated

1. gals/ft² of collector 1.52. $K_2 =$ (from Figure 11-5) = 1.07Load Heat Exchange Factor, K_4 3. ϵ_L (from Worksheet A, line 26) = 0.754. $C_{\text{hot water supply loop}} = \dot{m}c_p = C_H$
(from Worksheet A, lines 24 x 8.25 x 60) =6005 Btu/(hr·ft²)5. $C_{\text{air loop}} = \dot{m}c_p = C_A$
(from Worksheet A, line 25 x 0.075 x .24 x 60) =1296 Btu/(hr·ft²)6. $C_{\min} =$ smaller of C_H and $C_A =$ 1296 Btu/(hr·ft²)

7. (UA) bldg = (from Worksheet A, or B) =

714 Btu/(hr·°F)8. $\frac{\epsilon_L C_{\min}}{UA} =$ 1.369. $K_4 =$ (from Figure 11-6) =0.97

SYSTEM PERFORMANCE PARAMETERS X, Y

PROJECT Sunbody Residence
LOCATION _____

	1	2	3	4	5	6	7	8
Month	Tot. Monthly Radiation on Tilt Surf. S Btu/(Mo.·ft ²)	Total Heating Load L Btu/Mo. x 10 ⁶	S/L	Y F ₁ ·[3]	Mo. Av. Temp. T _a °F	212-T _a °F	Tot. Hrs in Mo. Δ time hr.	X
Jan.	44,609	21.5	.00207	.673	28	184	744	1.81
Feb.	42,924	18.0	.00238	.774	32	180	672	1.91
March	59,179	17.2	.00344	1.118	36	176	744	2.16
April	49,890	11.4	.00438	1.424	46	166	720	2.97
May	44,175	6.8	.00650	2.111	56	156	744	4.84
June	44,730	2.7	.0166	5.385	63	149	720	11.27
July	47,585	1.7	.0280	9.097	65	141	744	18.25
Aug.	45,198	1.8	.0251	8.161	65	147	744	17.23
Sept.	51,060	3.5	.0146	4.742	61	151	720	8.81
Oct.	51,429	9.0	.00571	1.856	51	161	744	3.77
Nov.	45,210	15.9	.00284	.923	38	174	720	2.23
Dec.	43,958	19.8	.00222	.722	32	180	744	1.92

- From Worksheet C, Col. 3
 - From Worksheet B, Col. 7
 - $$Y = \frac{A_c F'_1 (\tau \alpha) S}{C R'_L} \times K_4 = F_1 \cdot \frac{S}{L}$$
 - From Module 3
 - $$X = \frac{A_c F'_1 U_L (T_{ref} - T_a) \Delta time}{C R'_L} \times K_2$$

$$= F_2 \cdot \underbrace{([6] \cdot [7])}_{\text{columns}} \div [2]$$
- $$\frac{A_c}{F'_R} = \frac{500}{.67} \text{ ft}^2 \text{ (Wksht D)}$$

$$\frac{F'_1 U_L}{F'_R U_L} = \frac{.53}{.67} \text{ (Wksht D)}$$

$$K_2 = \text{ (Wksht E)}$$

$$K_4 = \text{ (Wksht E)}$$

$$F_1 = \frac{A_c F'_1 \tau \alpha}{C R'_L} K_4 = \text{ (Wksht E)}$$

$$F_2 = \frac{A_c F'_1 U_L}{C R'_L} K_2 = \text{ (Wksht E)}$$

FRACTION OF TOTAL HEATING LOAD SUPPLIED BY SOLAR ENERGY, F_{Annual}

PROJECT Sun body

COLLECTOR AREA 500 ft²
LOCATION Fr. Collins

Month	1 Tot. Mo. Htg. Load L Btu/mo.	2 System Parameter X	3 System Parameter Y	4 Solar Fraction/ mo. f	5 Actual Solar en/mo E Btu/mo.
Jan.	21.5	1.81	.67	.48	10.3
Feb.	18.0	1.91	.77	.53	9.5
March	17.2	2.16	1.12	.75	12.9
April	11.4	2.97	1.42	.86	9.8
May	6.8	4.84	2.11	1.0	6.8
June	2.7	11.27	5.39	1.0	2.7
July	1.7	18.25	9.10	1.0	1.7
Aug.	1.8	17.23	8.16	1.0	1.8
Sept.	3.5	8.81	4.74	1.0	3.5
Oct.	9.0	3.77	1.86	.95	8.6
Nov.	15.9	2.23	.92	.62	9.9
Dec.	19.8	1.92	.72	.50	9.9

$$L_{\text{tot}} = 129.3$$

$$E_{\text{tot}} = 87.4$$

$$F_{\text{Annual}} = \frac{E_{\text{tot}}}{L_{\text{tot}}} = \frac{87.4}{129.3} = 0.68$$

1. From Worksheet B
2. From Worksheet F, Column 8
3. From Worksheet F, Column 4
4. From "f chart", Figure 11-1
5. $E = f \times L$

SOLAR FRACTION FOR DIFFERENT COLLECTOR SIZES

In Example 11-2, the detailed calculations yielded the result that solar energy, with 500 ft² of RQP collectors and compatible system components, provides 68 percent of the total annual space and domestic hot water heating load for the building. To determine the annual solar fraction provided by a system with different collector areas, the f-chart used for the computations with 500 ft² of collectors, and Worksheet G are all that are needed. It is not necessary to rework the entire calculation procedure from Worksheet A through Worksheet G.

As an example, the monthly and annual solar fractions for 300 ft² of collectors can be determined by following the procedure below:

- Step 1. - For each month, locate the point (X,Y) on the f-chart that was calculated for 500 ft².
- Step 2. - Join the points with the origin, (X=0, Y=0), by a straight line.
- Step 3. - The distance from the origin to the point along the line represents 500 ft². Proportion the line length from the origin to the point and locate the point that represents 300 ft².
- Step 4. - Read the appropriate f-value representing 300 ft² and enter the value on Worksheet G (p. 11-36).
- Step 5. - Complete Worksheet G and determine the annual fraction, F.

The procedure described above is followed as shown in Figure 11-8. With 300 ft² of liquid-heating collectors, the system will provide 49 percent of the annual H and DHW load.

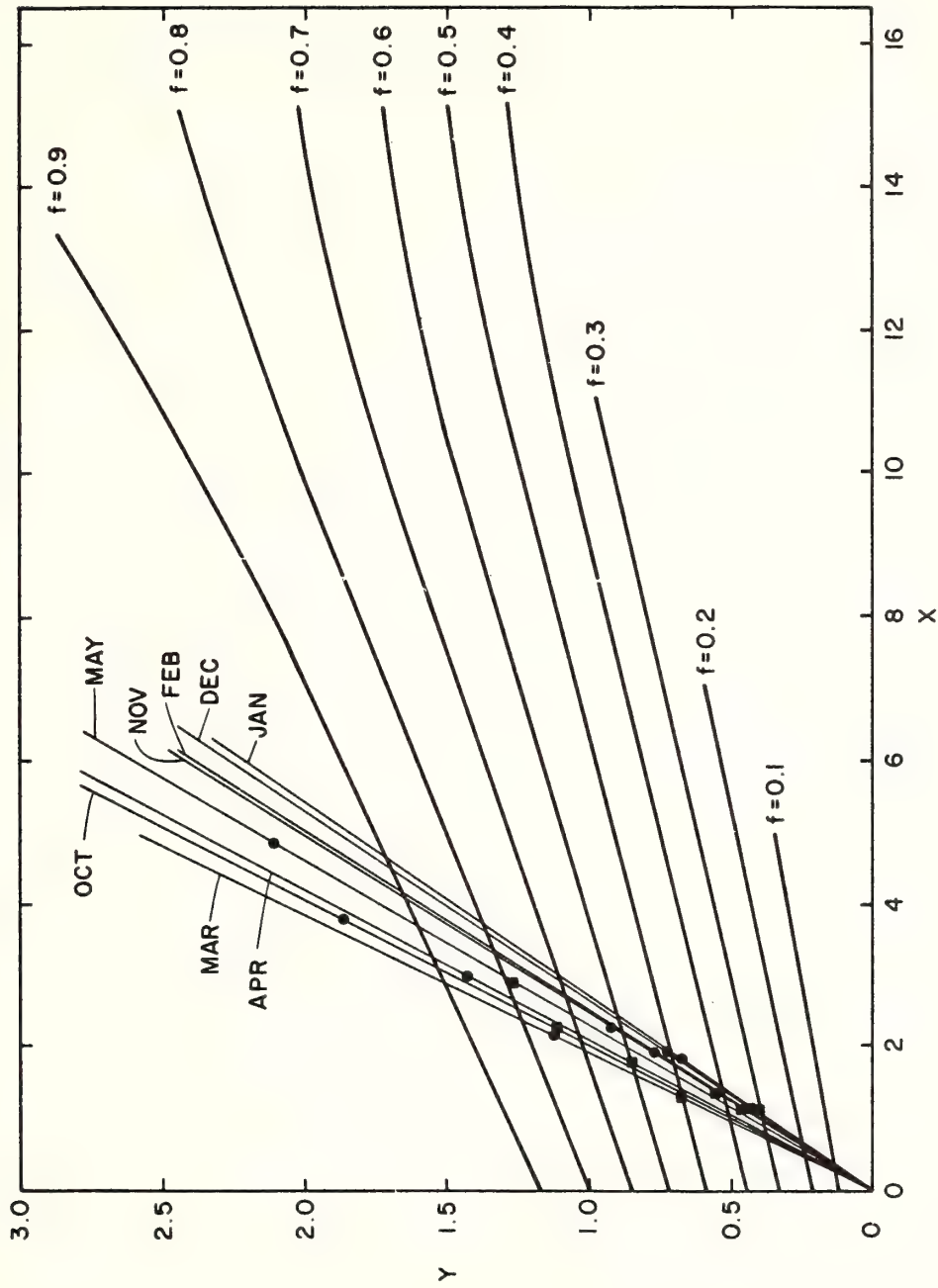


Figure 11 -8. f-Chart for Liquid-Heating Solar Systems
(for Example Problem 11-2)

FRACTION OF TOTAL HEATING LOAD SUPPLIED BY SOLAR ENERGY, F_{Annual}

PROJECT Synbody Residence

COLLECTOR AREA 300ft²
LOCATION FT. COLLINS

Month	1 Tot. Mo. Htg. Load L Btu/mo.	2 System Parameter X	3 System Parameter Y	4 Solar Fraction/ mo. f	5 Actual Solar en/mo E Btu/mo.
Jan.	21.5	1.09	.40	0.31	6.7
Feb.	18.0	1.15	.46	0.36	6.5
March	17.2	1.30	.67	0.50	8.6
April	11.4	1.78	.85	0.60	6.8
May	6.8	2.90	1.27	0.79	5.4
June	2.7	6.76	3.23	1.0	2.7
July	1.7	10.95	5.46	1.0	1.7
Aug.	1.8	10.34	4.90	1.0	1.8
Sept.	3.5	5.29	2.84	1.0	3.5
Oct.	9.0	2.26	1.12	0.74	6.7
Nov.	15.9	1.34	.55	0.42	6.7
Dec.	19.8	1.15	.43	0.33	6.5

$L_{\text{tot}} = 129.3$

$E_{\text{tot}} = 63.6$

$F_{\text{Annual}} = \frac{E_{\text{tot}}}{L_{\text{tot}}} = \frac{63.6}{129.3} = 0.49$

1. From Worksheet B
2. From Worksheet F, Column 8
3. From Worksheet F, Column 4
4. From "f chart", Figure 11-1
5. $E = f \times L$

Table 11-1

Monthly Temperature (T_m) in °F at Source for City Water in 14 Selected Cities

City	Source ¹	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
1. Phoenix	R _i , R _e , W	48	48	50	52	57	59	63	75	79	69	59	54
2. Miami	W	70	70	70	70	70	70	70	70	70	70	70	70
3. Los Angeles	R _i , W	50	50	54	63	68	73	74	76	75	69	61	55
4. Albuquerque	W	72	72	72	72	72	72	72	72	72	72	72	72
5. Las Vegas	W	73	73	73	73	73	73	73	73	73	73	73	73
6. Denver	Ri	39	40	43	49	55	60	63	64	63	56	45	37
7. Ft. Worth	L	56	49	57	70	75	81	79	83	81	72	56	46
8. Nashville	Ri	46	46	53	66	63	69	71	75	75	71	58	53
9. Washington, D.C.	Ri	42	42	52	56	63	67	67	78	79	68	55	46
10. Salt Lake City	W, C	35	37	38	41	43	47	53	52	48	43	38	37
11. Seattle	Ri	39	37	43	45	48	57	60	68	66	57	48	43
12. Boston	Re	32	36	39	52	58	71	74	67	60	56	48	45
13. Chicago	L	32	32	34	42	51	57	65	67	62	57	45	35
14. New York City	Re	36	35	36	39	47	54	58	60	61	57	48	45

¹Data from Handbook of Air Conditioning System Design, p. 5-41 through 5-46, McGraw Hill Book Company, New York (1965). Abbreviations: C-creek, L-lake, Re-reservoir, Ri-river, W-well.

SOLAR SYSTEM DATA

Building Owner _____

Address _____ Ph. _____

Contractor _____ Ph. _____

Type of System (liquid or air, H/DHW) _____

Site and Building Data

1. Location: Nearest City _____ Latitude _____
2. Building UA _____ Btu/(hr·°F)
3. DHW volume per day _____ gallons/day
4. Collector manufacturer _____
5. Collector area _____ ft²
6. Collector tilt _____ degrees
7. Tilt = latitude + _____ degrees
8. Collector orientation _____ degrees _____ from south
9. Collector shading _____ % in December
10. Collector efficiency data

(a) $F_R(\tau\alpha)_n$ _____

(b) $F_R U_L$ _____ Btu/(hr·ft²·°F)

(c) Fluid temperature basis (circle one)

Case 1 T_i

Case 2 $\frac{T_i + T_{out}}{2}$

Case 3 T_{out}

11. Collector Fluid:

(a) Composition: _____

(b) Flow rate G _____ ft³/min

(c) Capacitance Flow rate $\frac{(\text{cfm}) \times (\text{density}) \times (\text{specific heat}) \times 60 \text{ m/hr}}{\text{collector area}}$
= _____ Btu/(hr·ft²·°F)

Storage Data

12. Storage medium _____

13. Unit volume _____ ft³/ft²

14. Total volume (item 5 x 13) _____ ft³

15. Storage Capacitance $\frac{(\text{vol}) \times (\text{density}) \times (\text{sp. heat})}{\text{collector area}}$
_____ Btu/(ft²·°F)

Auxiliary Furnace/Boiler

Type _____

Manufacturer _____

Rated Capacity _____ Btu/hr

Auxiliary energy source _____

Auxiliary DHW Unit

Size _____ gal

Auxiliary energy source _____

Hot water set temperature _____ °F

HEATING AND/OR DOMESTIC HOT WATER LOAD, L

Project _____

Month	1 Monthly Degree Days DD °F·days	2 Monthly Space Htg Load Q_s Btu/Mo.	No. of Days/ Mo. N	3 Vol. of DHW Used/Mo. Gal./Mo.	4 Temp. Water Main Sup. T_m °F	5 DHW Temp. Rise $T_{HW}-T_m$ °F	6 Monthly DHW Load Q_w Btu/Mo.	7 Total Heating Load L Btu/Mo.
Jan.			31					
Feb.			28					
March			31					
April			30					
May			31					
June			30					
July			31					
Aug.			31					
Sept.			30					
Oct.			31					
Nov.			30					
Dec.			31					
Totals								

$Q_d =$ _____ Btu/h
(Given data or calculate
as in Module 5)

DTD = $70 - T_o$
= $70 -$ _____ =

Where: T_o = 99% winter
design temperature.
(From ASHRAE Fundamentals,
or Table A5-10)

70°F = indoor design
temperature

$UA = \frac{Q_d}{DTD} =$

$T_{HW} =$ _____

- From Table A5-10 or Figures A5-1 through A5-12
- $Q_s = (24)(UA)(\text{Degree Day})$
- $(\text{Vol/day}) \times (\text{no. days/mo.}) =$ _____ (gal./day) \times (no. days/mo.)
- From Table 11-1 for selected cities
- $Q_w = (\text{vol. of water}) \times 8.34 \times 1 \times (T_{HW} - T_m)$
- $L = Q_s + Q_w$

TOTAL MONTHLY SOLAR RADIATION AVAILABLE, S

Project _____

Location _____

Collector Tilt _____

Nearest Data Site _____

	1	2	3
Month	Monthly Avg. Daily Rad. on Tilt Surf. \bar{I}_T Btu/(Day·ft ²)	No. of Days in month N	Tot. Monthly Radiation on Tilt Surf. S Btu/(Mo·ft ²)
Jan.		31	
Feb.		28	
March		31	
April		30	
May		31	
June		30	
July		31	
Aug.		31	
Sept.		30	
Oct.		31	
Nov.		30	
Dec.		31	

COLLECTOR COMBINED PERFORMANCE CHARACTERISTICS, $F_R(\overline{\tau\alpha})$, $F_R U_L$

PROJECT _____

Collector Efficiency Data from Worksheet A (lines 10(a), (b))

1. Intercept, $F_R(\tau\alpha)_n$ = _____

2. Slope, $F_R U_L$ = _____

Reference Temperature Basis: 1. t_{in} , 2. $\frac{t_{in} + t_{out}}{2}$, 3. t_{out}

3. Collector area, A_c = _____ ft^2

4. Collector volumetric flow rate (Worksheet A, 11(d))
_____ ft^3/min

Correction to t_{in} basis

5. Case 1: (no correction) $F_R(\tau\alpha)_n$ = _____
 $F_R U_L$ = _____

6. Case 2: $F_R(\tau\alpha)_n = F_R \tau\alpha \times \frac{1}{1 + \frac{F_R U_L A_c}{2 C_c}} = \left[\text{_____} \right]$
 $F_R U_L = F_R U_L \times \frac{1}{1 + \frac{F_R U_L A_c}{2 C_c}} = \left[\text{_____} \right]$

$C_c = \dot{m}_{cp} = (\text{volumetric flow rate})(\text{density})(\text{time conversion})(\text{specific heat})$

where: for liquids, density = (8.34 lb/gal) x
(specific gravity) for air, density = 0.075 lb/ft³
at 70° and 1 atm. specific heat = 0.24 Btu/lb·°F

7. Case 3: $F_R(\tau\alpha)_n = F_R(\tau\alpha)_n \times \frac{1}{1 + \frac{F_R U_L A_c}{C_c}} = \left[\text{_____} \right]$
 $F_R U_L = F_R U_L \times \frac{1}{1 + \frac{F_R U_L A_c}{C_c}} = \left[\text{_____} \right]$

8. Incident Angle Modifier, $\frac{(\overline{\tau\alpha})}{(\tau\alpha)_n} = \begin{cases} .91, & \text{for two cover plates} \\ .93, & \text{for one cover plate} \end{cases}$

9. $F_R(\overline{\tau\alpha}) = F_R(\tau\alpha)_n \times \frac{\overline{\tau\alpha}}{(\tau\alpha)_n} = \underline{0.69} \times \underline{0.93} = \underline{0.64}$

CORRECTION FACTORS, K_1 , K_2

PROJECT _____

Collector Flow Factor, K_1

1. Air Flow rate (Worksheet A, line 11(b)) = _____ cfm
2. Capacitance Flow rate (from Worksheet A,
line 11(c)) _____ Btu/(h·ft²·°F)
3. K_1 = (from Figure 11-4) = _____

Storage Mass Capacitance Factor, K_2

4. Storage capacitance (Worksheet A,
line 15 =) _____ Btu/ft²·°F
5. K_2 = (from Figure 11-5) = _____

SYSTEM PERFORMANCE PARAMETERS X, Y

PROJECT _____

LOCATION _____

	1	2	3	4	5	6	7	8
	Tot. Monthly Radiation on Tilt Surf. S Btu/(Mo·ft ²)	Total Heating Load L Btu/Mo.		Y $F_1 \cdot [3]$	Mo. Av. Temp. T_a °F	212- T_a °F	Tot. Hrs in Mo. Δ time hr.	
Month			S/L					X
Jan.							744	
Feb.							672	
March							744	
April							720	
May							744	
June							720	
July							744	
Aug.							744	
Sept.							720	
Oct.							744	
Nov.							720	
Dec.							744	

1. From Worksheet C, Col. 3

2. From Worksheet B, Col. 7

4. $Y = \frac{A_c F_R (\tau \alpha) S}{L} = F_1 \cdot \frac{S}{L}$

5. From Module 3 :

8. $X = \frac{A_c F_R U_L (T_{ref} - T_a) \Delta time}{L} \times K_1 \times K_2$

$= F_2 \cdot [(6) \cdot (7)] : [2]$

$A_c \frac{F_R}{\tau \alpha} =$ _____ ft²
 $F_R U_L =$ _____ (Wksht D)
 $F_R U_L =$ _____ (Wksht D)
 $K_1 =$ _____ (Wksht E)
 $K_2 =$ _____ (Wksht E)
 $F_1 = A_c F_R \tau \alpha =$ _____
 $F_2 = A_c F_R U_L K_1 K_2 =$ _____

FRACTION OF TOTAL HEATING LOAD SUPPLIED BY SOLAR ENERGY, F_{Annual}

PROJECT _____

	1	2	3	4	5
Month	Tot. Mo. Htg. Load $L \times 10^6$ Btu/mo.	System Parameter X	System Parameter Y	Solar Fraction/ mo. f	Actual Solar en/mo $E \times 10^6$ Btu/mo.
Jan.					
Feb.					
March					
April					
May					
June					
July					
Aug.					
Sept.					
Oct.					
Nov.					
Dec.					

COLLECTOR AREA _____ ft^2
LOCATION _____

$L_{\text{tot}} =$

$E_{\text{tot}} =$

$$F_{\text{Annual}} = \frac{E_{\text{tot}}}{L_{\text{tot}}} = \frac{\quad}{\quad} = \frac{\quad}{\quad}$$

1. From Worksheet B
2. From Worksheet F, Column 8
3. From Worksheet F, Column 4
4. From "f chart", Figure 11-2 or Equation (11-4)
5. $E = f \times L$

SOLAR SYSTEM DATA

Building Owner _____

Address _____ Ph. _____

Contractor _____ Ph. _____

Type of System (liquid or air, H/DHW) _____

Site and Building Data

1. Location: Nearest City _____ Latitude _____
2. Building UA _____ Btu/(hr·°F)
3. DHW volume per day _____ gallons/day
4. Collector manufacturer _____
5. Collector area _____ ft²
6. Collector tilt _____ degrees
7. Tilt = latitude + _____ degrees
8. Collector orientation _____ degrees _____ from south
9. Collector shading _____ % in December
10. Collector efficiency data
 - (a) $F_R(\tau\alpha)_n$ _____
 - (b) $F_R U_L$ _____ Btu/(hr·ft²·°F)
 - (c) Fluid temperature basis (circle one)

Case 1	T_i
Case 2	$\frac{T_i + T_{out}}{2}$
Case 3	T_{out}
11. Collector Fluid:
 - (a) Composition: _____
 - (b) Specific heat, c_p (from Mod. 4) _____ Btu/(lb·°F)
 - (c) Fluid density, ρ (from Mod. 4) _____ lb/gal
 - (d) Flow rate G _____ gal/min

Storage Data

12. Storage medium _____
13. Unit volume _____ gal/ft² or ft³/ft²
14. Total volume (item 5 x 13) _____ gal or ft³

15. Specific heat of storage material c_p _____ Btu/(lb·°F)
16. Density _____ lb/gal
17. Total mass (14 x 16) M _____ lb.
18. Total heat capacity (17 x 15)
 $C_s = Mc_p$ _____ Btu/°F
19. Total heat capacity per unit
collector area (18 ÷ 5) _____ Btu/(ft²·°F)

Heat Exchangers

20. Collector/storage type _____ and
manufacturer _____
21. Storage loop flow rate _____ gal/min
22. Heat exchange effectiveness ϵ_{cs} _____
23. Load heat exchanger type _____ and
manufacturer _____
24. Load loop flow rate _____ gal/min
25. Building air supply flow rate _____ ft³/min
26. Heat exchanger effectiveness ϵ_L _____

DHW Pre-heater

27. Collector/storage heat exchanger type _____ and
manufacturer _____
28. Collector loop flow rate _____ gal/min
29. Heat exchanger effectiveness ϵ_{Hw} _____
30. Storage volume _____ gal
31. Storage mass, $M_{St.}$ (line 30 x 8.34) _____ lb

Auxiliary Furnace/Boiler

32. Type _____
33. Manufacturer _____
34. Rated capacity _____ Btu/hr
35. Auxiliary energy source _____

Auxiliary DHW Unit

36. Size _____ gal
37. Auxiliary energy source _____
38. Hot water set temperature _____ °F

HEATING AND/OR DOMESTIC HOT WATER LOAD, L

Project _____

	1	2		3	4	5	6	7
Month	Monthly Degree Days DD °F-days	Monthly Space Htg Load Q_s Btu/Mo.	No. of Days/Mo. N	Vol. of DHW Used/Mo. Gal./Mo.	Temp. Water Main Sup T_m °F	DHW Temp. Rise $T_{HW} - T_m$ °F	Monthly DHW Load Q_w Btu/Mo.	Total Heating Load L Btu/Mo.
Jan.			31					
Feb.			28					
March			31					
April			30					
May			31					
June			30					
July			31					
Aug.			31					
Sept.			30					
Oct.			31					
Nov.			30					
Dec.			31					
						Total		

1. From Table A5-10 or Figures A5-1 through A5-12

2. $Q_s = (24)(UA)(\text{Degree Day})$

3. $(\text{Vol/day}) \times (\text{no. days/mo.}) = \text{_____} (\text{gal./day}) \times (\text{no. days/mo.})$

4. From Table 11-1 for selected cities.

6. $Q_w = (\text{vol. of water}) \times 8.34 \times 1 \times (T_{HW} - T_m)$

7. $L = Q_s + Q_w$

$Q_d = \text{_____ Btu/h}$
(Given data or calculate as in Module 5)

$DTD = 70 - T_o$
 $= 70 - \text{_____} = \text{_____}$

Where: $T_o = 99\%$ winter design temperature.
(From ASHRAE Fundamentals, or Table A5-10)

$70^\circ\text{F} = \text{indoor design temperature}$

$UA = \frac{Q_d}{DTD} = \text{_____}$

$T_{HW} = \text{_____}$

TOTAL MONTHLY SOLAR RADIATION AVAILABLE, S

Project _____

Location _____

Collector Tilt _____

Nearest Data Site _____

	1	2	3
Month	Monthly Avg. Daily Rad. on Tilt Surf. \bar{I}_T Btu/(Day·ft ²)	No. of Days in month N	Tot. Monthly Radiation on Tilt Surf. S Btu/(Mo·ft ²)
Jan.		31	
Feb.		28	
March		31	
April		30	
May		31	
June		30	
July		31	
Aug.		31	
Sept.		30	
Oct.		31	
Nov.		30	
Dec.		31	

COLLECTOR COMBINED PERFORMANCE CHARACTERISTICS, $F_R'(\overline{\tau\alpha}) \cdot F_R' U_L$

PROJECT _____

Collector Efficiency Data from Worksheet A (lines 10(a), (b))

1. Intercept, $F_R(\tau\alpha)_n$ = _____

2. Slope, $F_R U_L$ = _____

Reference Temperature Basis: 1. t_{in} , 2. $\frac{t_{in} + t_{out}}{2}$, 3. t_{out}

3. Collector area, A_c = _____ ft^2

4. Collector volumetric flow rate (Worksheet A, 11(d))
 _____ gal/min or ft^3/min

Correction to t_{in} basis

5. Case 1: (no correction) $F_R(\tau\alpha)_n$ = _____
 $F_R U_L$ = _____

6. Case 2: $F_R(\tau\alpha)_n = F_R \tau\alpha \times \left[\frac{1}{1 + \frac{F_R U_L A_c}{2 C_c}} \right] =$ _____

$F_R U_L = F_R U_L \times \left[\frac{1}{1 + \frac{F_R U_L A_c}{2 C_c}} \right] =$ _____

$C_c = \dot{m} c_p = (\text{volumetric flow rate})(\text{density})(\text{time conversion})(\text{specific heat})$
 = _____ = _____

where: for liquids, density = (8.34 lb/gal) x
 (specific gravity) for air, density = 0.075 lb/ ft^3
 at 70° and 1 atm. specific heat = 0.24 Btu/lb·°F

7. Case 3: $F_R(\tau\alpha)_n = F_R(\tau\alpha)_n \times \left[\frac{1}{1 + \frac{F_R U_L A_c}{C_c}} \right] =$ _____

$F_R U_L = F_R U_L \times \left[\frac{1}{1 + \frac{F_R U_L A_c}{C_c}} \right] =$ _____

8. Incident Angle Modifier, $\frac{(\overline{\tau\alpha})}{(\tau\alpha)_n} = \begin{cases} .91 & \text{for two cover plates} \\ .93 & \text{for one cover plate} \end{cases}$

Collector Loop Heat Exchanger Modifier, $\frac{F'_R}{F_R}$

9. For air systems and liquid systems without a collector/storage heat exchanger, $\frac{F'_R}{F_R} = 1$

Capacitance Rate:

10. $C_c =$ (from line 6) = _____ Btu/(hr·°F)
11. $C_s =$ (calc. as for C_c above) = _____ Btu/(hr·°F)
12. $C_{\min} =$ (lesser of C_c and C_s) = _____ Btu/(hr·°F)
13. Collector Storage Heat Exchanger Effectiveness, $\epsilon_{cs} =$ _____
14. $x = \frac{C_c}{\epsilon_{cs} C_{\min}} =$ (from Worksheet A, line 22) = _____
15. $y = \frac{A_c (F_R U_L)}{C_c} =$ _____ = _____
16. $\frac{F'_R}{F_R} = \frac{1}{1 + y(x-1)} =$ _____
17. $F'_R(\overline{\tau\alpha}) = F_R(\tau\alpha)_n \times \frac{(\overline{\tau\alpha})}{(\tau\alpha)_n} \times \frac{F'_R}{F_R} =$ _____
18. $F'_R U_L = F_R U_L \times \frac{F'_R}{F_R} =$ _____

CORRECTION FACTORS, K_2 , K_4

PROJECT _____

Storage Mass Capacitance Factor, K_2

Note: M includes hot water storage volume where it is solar heated

1. gals/ft² of collector _____
2. K_2 = (from Figure 11-5) = _____

Load Heat Exchange Factor, K_4

3. ϵ_L (from Worksheet A, line 26) = _____
4. $C_{\text{hot water supply loop}} = \dot{m}c_p = C_H$
(from Worksheet A, lines 24 x 8.25 x 60) = _____ Btu/(hr·ft²)
5. $C_{\text{air loop}} = \dot{m}c_p = C_A$
(from Worksheet A, line 25 x 0.075 x .24 x 60) = _____ Btu/(hr·ft²)
6. C_{\min} = smaller of C_H and C_A = _____ Btu/(hr·ft²)
7. (UA) bldg = (from Worksheet A, or B) = _____ Btu/(hr·°F)
8. $\frac{\epsilon_L C_{\min}}{UA} =$ _____
9. K_4 = (from Figure 11-6) = _____

SYSTEM PERFORMANCE PARAMETERS X, Y

PROJECT _____
LOCATION _____

	1	2	3	4	5	6	7	8
Month	Tot. Monthly Radiation on Tilt Surf. S Btu/(Mo.·ft ²)	Total Heating Load L Btu/Mo.	S/L	Y F ₁ ·[3]	Mo. Av. Temp. T _a °F	212-T _a °F	Tot. Hrs in Mo. Δ time hr.	X
Jan.							744	
Feb.							672	
March							744	
April							720	
May							744	
June							720	
July							744	
Aug.							744	
Sept.							720	
Oct.							744	
Nov.							720	
Dec.							744	

1. From Worksheet C, Col. 3

2. From Worksheet B, Col. 7

4. $Y = \frac{A_c F'_R (\overline{\tau\alpha}) S}{L} \times K_4 = F_1 \cdot \frac{S}{L}$

5. From Module 3

8. $X = \frac{A_c F'_R U_L (T_{ref} - T_a) \Delta time}{L} \times K_2$

$= F_2 \cdot \underbrace{([6] \cdot [7])}_{\text{columns}} [2]$

$A_c \overline{\tau\alpha} =$

$F'_R \overline{\tau\alpha} =$

$F'_R U_L =$

$K_2 =$

$K_4 =$

$F_1 = A_c F'_R \overline{\tau\alpha}$

$F_2 = A_c F'_R U_L$

ft^2

(Wksht D)

(Wksht D)

(Wksht E)

(Wksht E)

FRACTION OF TOTAL HEATING LOAD SUPPLIED BY SOLAR ENERGY, F_{Annual}

PROJECT _____

	1	2	3	4	5
Month	Tot. Mo. Htg. Load L Btu/mo.	System Parameter X	System Parameter Y	Solar Fraction/ mo. f	Actual Solar en/mo E Btu/mo.
Jan.					
Feb.					
March					
April					
May					
June					
July					
Aug.					
Sept.					
Oct.					
Nov.					
Dec.					

COLLECTOR AREA _____ ft^2
LOCATION _____

$L_{\text{tot}} =$

$E_{\text{tot}} =$

$$F_{\text{Annual}} = \frac{E_{\text{tot}}}{L_{\text{tot}}} = \frac{\quad}{\quad} = \frac{\quad}{\quad}$$

1. From Worksheet B
2. From Worksheet F, Column 8
3. From Worksheet F, Column 4
4. From "f chart", Figure 11-1
5. $E = f \times L$

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 12

ECONOMIC CONSIDERATIONS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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OBJECTIVES

The objectives of this module are to describe methods for determining solar heating costs on a life-cycle basis and on a cash-flow basis and for comparing these costs with those of non-solar systems. The participant of this workshop should be able to:

1. Estimate the installed cost of a solar system.
2. Calculate the total life-cycle cost of a solar heating system.
3. Estimate the economic feasibility of a solar system.

INTRODUCTION

Solar heating systems require higher capital costs than conventional systems, and economic evaluations invariably involve cost comparisons between the two systems. Comparisons which do not account for future fuel cost savings for heating are both misleading and unfavorable to solar systems. By accounting for capital and operating costs of heating alternatives over a period of time, deemed to be the "life" of the systems, the relative economic merits of "paying for hardware" or "paying for energy" can be assessed. Thus, life-cycle costing methods have generally been used to make economic comparisons, although speculative assumptions for the rates of increase in energy prices, discount rates, costs for goods and services, property tax, and insurance must be used in the analysis. Credit is taken for income tax deductions allowable for property tax payments and mortgage (loan) interest payments. While life-cycle cost analysis is a fair method for

comparing solar and non-solar systems, consider annual cash flow differences between the two types of heating systems. Both methods are explained in this module.

The economic principles in this module have been presented in sufficient detail for verification by financial specialists. Much of this material, such as the development of equations for discounted cash flow, need not be learned by the trainee. By following the examples in the work sheets and using the charts and tables in this module, calculations are simplified. Solar heating practitioners concerned mainly with selecting and installing solar heating systems may seldom need economic information other than the costs of equipment and installation labor. Other material in this module can be, however, useful and important for those who are responsible for making decisions and providing recommendations on the most economical methods for the heating of buildings.

FACTORS IN ANALYSES

The total cost of a solar heating system, over its useful life, includes (1) capital and installation costs or mortgage payments on money borrowed to pay for the installed system, (2) fuel cost for the auxiliary unit, (3) operating and maintenance (O and M) costs, and (4) income tax credits for initial investment and for annual interest and property tax payments. With a non-solar heating system the capital and O and M costs are small but fuel costs are high (and rising steadily). A solar system has large capital costs, lower fuel costs, and non-negligible O and M costs. A comparison is necessary to determine whether a solar system is economical compared to a non-solar system, and

the comparison is usually made for a selected number of years, which is estimated to be the "life" of a system. A life-cycle cost analysis is first explained followed by an annual cash-flow analysis.

The yearly cash flow for a residential solar heating system is:

$$\begin{aligned} \text{Yearly cost with solar} = & \text{Mortgage payment} + \text{Auxiliary fuel cost} + \text{Property tax increase} + \text{Insurance premium} \\ & + \text{Operating costs} + \text{Maintenance cost} - \text{Income tax savings for interest and taxes paid} \end{aligned} \quad (12-1)$$

whereas for a non-solar system,

$$\text{Yearly cost for non-solar} = \text{Fuel cost} + \text{Operating and maintenance costs} \quad (12-2)$$

In commercial buildings there are other factors such as depreciation of equipment and salvage value to be considered.

The sum of the yearly cash flows over the "life" of the system can be construed as the life-cycle cost of the system, and the costs of the solar and non-solar systems can be compared over an equal life-time of n years, to determine which system would be more expensive. Cash flow calculations should include inflation, with fuel costs probably increasing more rapidly than costs of general goods and services (at least in the near-term). The use of different inflation factors for the items in Equation (12-1) or (12-2) in effect gives more weight to some cost items than to others, say fuel costs over mortgage payments as an example, particularly if mortgage payment is fixed and fuel cost rises.

Because the sum of annual cash flows for both solar and non-solar systems would in effect add different value dollars each year as a consequence of inflation, a more appropriate economic comparison is made on the basis of present worth, which discounts future expenditures

to the value of first year dollars. Hence, when the present worth of future annual expenditure is computed, and these values are added, equivalent-value dollars are being considered. When the inflation and discount factors are taken into consideration in a life-cycle cost analysis, Equations (12-1) and (12-2) may be rewritten as follows:

$$C_T(\text{solar}) = (AC_a)E_1 + C_oE_o + C_mE_m + (1-F)Lc_fE_f \quad (12-3)$$

and

$$C_{TC}(\text{non-solar}) = C_{oc}E_o + C_{mc}E_m + Lc_fE_f \quad (12-4)$$

where

A is the collector area, ft^2 ,

C_T is the total life-cycle cost of the solar system, \$,

C_{TC} is the total life-cycle cost of the non-solar system, \$,

C_a is the installed cost of the solar system per unit collector area, $\$/\text{ft}^2$,

C_o is the first year operating cost for the solar system, $\$/\text{yr}$,

C_{oc} is the first year operating cost for the non-solar system, $\$/\text{yr}$,

C_m is the first year maintenance cost for the solar system, $\$/\text{yr}$,

C_{mc} is the first year maintenance cost for the non-solar system, $\$/\text{yr}$,

c_f is the first year fuel cost per unit of delivered heat, $\$/\text{MMBtu}$,

E_1 is an economic factor which accounts for downpayment, mortgage interest rate, insurance rate, property tax rate, income tax saving, inflation rate, and market discount rate,

E_o is an economic factor which accounts for inflation rate of operating cost and market discount rate,

E_m is an economic factor which accounts for inflation rate of maintenance cost and market discount rate,

E_f is an economic factor which accounts for fuel inflation rate and market discount rate,

F is the fraction of annual heat provided by the solar system,

L is the annual heating load for the building, MMBtu.

The economic factors, E_o , E_m , and E_f are the sums of annual compounded inflation factors discounted annually to present worth. The present worth of the sum of an annuity over a life-time of n years inflated at a constant rate and discounted at a constant rate can be written as:

$$P/X(d,r,n) = \frac{(1+d)^n - (1+r)^n}{(1+d)^n(d-r)} \text{ for } d \neq r \quad (12-5)$$

and

$$P/X(d,r,n) = n/(1+r) \text{ for } d = r \quad (12-6)$$

where P is the present value of an annuity over n years,

X is the first year cost,

d is the discount rate,

r is the inflation rate,

n is the years of analysis or life of the system.

The notation (d,r,n) after P/X indicates that the value of P/X refers to values of d , r , and n , placed in the appropriate terms in Equations (12-5) and (12-6). Tables of P/X values are provided in this module for an appropriate range of d , r , and n in Tables 12-1 through 12-6.

The economic factors can now be expressed as:

$$E_o = P/X(d,r_o,n) \text{ years,} \quad (12-7)$$

$$E_m = P/X(d,r_m,n) \text{ years,} \quad (12-8)$$

$$E_f = P/X(d,r_f,n) \text{ years.} \quad (12-9)$$

The economic factor E_1 is slightly more involved and is expressed as:

Table 12-1

Values of P/X (d, r, n) for Discount Rate of 0 Percent

Years	Annual Rate of Increase					
	0	3	6	8	10	12
10	10.0	11.464	13.181	14.487	15.937	17.549
11	11.0	12.808	14.972	16.645	18.531	20.655
12	12.0	14.192	16.870	18.977	21.384	24.133
13	13.0	15.618	18.882	21.495	24.523	28.029
14	14.0	17.086	21.015	24.215	27.975	32.393
15	15.0	18.599	23.276	27.152	31.772	37.280
16	16.0	20.157	25.673	30.324	35.950	42.753
17	17.0	21.762	28.213	33.750	40.545	48.884
18	18.0	23.414	30.906	37.450	45.599	55.750
19	19.0	25.117	33.760	41.446	51.159	63.440
20	20.0	26.870	36.786	45.762	57.275	72.052
21	21.0	28.676	39.993	50.423	64.002	81.669
22	22.0	30.537	43.392	55.457	71.403	92.503
23	23.0	32.453	46.996	60.893	79.543	104.603
24	24.0	34.426	50.816	66.765	88.497	188.155
25	25.0	36.459	54.865	73.106	98.347	133.334
26	26.0	38.553	59.156	79.954	109.182	150.334
27	27.0	40.710	63.706	87.351	121.100	169.374
28	28.0	42.931	68.528	95.339	134.210	190.699
29	29.0	45.219	73.640	103.966	148.631	214.583
30	30.0	47.575	79.058	113.283	164.494	241.333

Table 12-2

Values of P/X (d , r , n) for Discount Rate of 4 Percent

Years	Annual Rate of Increase					
	0	3	6	8	10	12
10	8.111	9.210	10.492	11.462	12.537	13.727
11	8.760	10.083	11.655	12.865	14.222	15.845
12	9.385	10.947	12.841	14.321	16.004	17.918
13	9.986	11.804	14.049	15.833	17.889	20.258
14	10.563	12.652	15.281	17.404	19.883	22.777
15	11.118	13.492	16.536	19.035	21.991	25.491
16	11.652	14.323	17.816	20.728	24.222	28.413
17	12.166	15.147	19.120	22.487	26.580	31.561
18	12.659	15.963	20.449	24.314	29.076	34.950
19	13.134	16.771	21.804	26.210	31.714	38.600
20	13.590	17.571	23.185	28.180	34.506	42.531
21	14.029	18.364	24.592	30.225	37.458	46.764
22	14.451	19.149	26.027	32.349	40.581	51.322
23	14.857	19.926	27.489	34.555	43.883	56.232
24	15.247	20.696	28.979	36.846	47.377	61.519
25	15.622	21.459	30.498	39.224	51.071	67.213
26	15.983	22.214	32.046	41.695	54.979	73.344
27	16.330	22.962	33.623	44.260	59.113	79.448
28	16.663	23.703	35.232	46.924	63.485	87.059
29	16.984	24.436	36.871	49.690	69.109	94.718
30	17.292	25.163	38.541	52.563	73.000	102.965

Table 12-3

Values of P/X (d , r , n) for Discount Rate of 6 Percent

Years	Annual Rate of Increase					
	0	3	6	8	10	12
10	7.360	8.319	9.434	10.277	11.208	12.238
11	7.887	9.027	10.377	11.414	12.575	13.874
12	8.384	9.715	11.321	12.573	13.993	15.603
13	8.853	10.383	12.264	13.753	15.464	17.430
14	9.295	11.033	13.208	14.956	16.991	19.360
15	9.712	11.664	14.151	16.182	18.575	21.399
16	10.106	12.277	15.094	17.430	20.220	23.553
17	10.477	12.873	16.038	18.703	21.926	25.830
18	10.828	13.452	16.981	19.999	23.697	28.236
19	11.158	14.015	17.925	21.320	25.535	30.777
20	11.470	14.562	18.868	22.665	27.442	33.463
21	11.764	15.093	19.811	24.036	29.421	36.300
22	12.042	15.609	20.755	25.433	31.474	39.298
23	12.303	16.111	21.698	26.857	33.605	42.466
24	12.550	16.598	22.642	28.307	35.817	45.813
25	12.783	17.072	23.585	29.784	38.112	49.350
26	13.003	17.532	24.528	31.290	40.493	53.087
27	13.211	17.979	25.472	32.823	42.965	57.035
28	13.406	18.414	26.415	34.386	45.530	61.207
29	13.591	18.836	27.358	35.978	48.191	65.615
30	13.765	19.246	28.302	37.601	50.953	70.272

Table 12-4

Values of P/X (d , r , n) for Discount Rate of 8 Percent

Years	Annual Rate of Increase					
	0	3	6	8	10	12
10	6.710	7.550	8.525	9.259	10.070	10.965
11	7.139	8.127	9.293	10.185	11.183	12.297
12	7.536	8.676	10.046	11.111	12.316	13.679
13	7.904	9.200	10.786	12.037	13.470	15.111
14	8.244	9.700	11.513	12.963	14.645	16.597
15	8.559	10.177	12.225	13.889	15.842	18.137
16	8.851	10.632	12.926	14.815	17.061	19.735
17	9.122	11.066	13.611	15.741	18.303	21.392
18	9.372	11.479	14.285	16.667	19.568	23.110
19	9.604	11.874	14.947	17.593	20.856	24.892
20	9.818	12.250	15.596	18.519	22.169	26.740
21	10.017	12.609	16.233	19.444	23.505	28.656
22	10.201	12.951	16.858	20.370	24.866	30.643
23	10.371	13.277	17.472	21.296	26.253	32.704
24	10.529	13.589	18.074	22.222	27.665	34.841
25	10.675	13.885	18.666	23.148	29.103	37.058
26	10.810	14.169	19.246	24.074	30.568	39.356
27	10.935	14.438	19.815	25.000	32.060	41.740
28	11.051	14.696	20.374	35.926	33.580	44.212
29	11.158	14.942	20.923	26.852	35.127	46.775
30	11.258	15.176	21.461	27.778	36.704	49.433

Table 12-5

Values of P/X (d, r, n) for Discount Rate of 10 Percent

Years	Annual Rate of Increase					
	0	3	6	8	10	12
10	6.145	6.884	7.739	8.382	9.091	9.872
11	6.495	7.355	8.366	9.139	10.000	10.961
12	6.814	7.796	8.971	9.882	10.909	12.069
13	7.103	8.209	9.554	10.611	11.818	13.197
14	7.367	8.596	10.116	11.377	12.727	14.346
15	7.606	8.958	10.657	12.030	13.636	15.516
16	7.824	9.297	11.179	12.721	14.545	16.708
17	8.022	9.614	11.681	13.399	15.455	17.920
18	8.201	9.911	12.166	14.064	16.365	19.155
19	8.365	10.190	12.632	14.717	17.273	20.413
20	8.514	10.450	13.082	15.359	18.182	21.693
21	8.649	10.695	13.515	15.989	19.091	22.997
22	8.772	10.923	13.933	16.607	20.000	24.324
23	8.883	11.137	14.335	17.214	20.909	25.675
24	8.985	11.337	14.723	17.810	21.818	27.051
25	9.077	11.525	15.097	18.396	22.727	28.452
26	9.161	11.701	15.457	18.970	23.636	29.878
27	9.237	11.865	15.804	19.534	24.545	31.331
28	9.307	12.019	16.138	20.088	25.455	32.809
29	9.370	12.613	16.461	20.632	26.364	34.315
30	9.427	12.299	16.771	21.166	27.273	35.848

Table 12-6

Values of P/X (d , r , n) for Discount Rate of 12 Percent

Years	Annual Rate of Increase					
	0	3	6	8	10	12
10	5.650	6.303	7.057	7.822	8.244	8.929
11	5.938	6.690	7.571	8.243	8.990	9.821
12	6.194	7.045	8.059	8.841	9.722	10.714
13	6.424	7.372	8.520	9.418	10.441	11.607
14	6.628	7.672	8.956	9.975	11.148	12.500
15	6.811	7.949	9.369	10.511	11.842	13.393
16	6.974	8.203	9.760	11.029	12.523	14.286
17	7.120	8.436	10.130	11.528	13.192	15.179
18	7.250	8.651	10.480	12.009	13.850	16.071
19	7.366	8.849	10.812	12.473	14.495	16.964
20	7.469	9.031	11.125	12.920	15.129	17.857
21	7.562	9.198	11.422	13.352	15.752	18.758
22	7.645	9.352	11.703	13.768	16.363	19.643
23	7.718	9.493	11.969	14.169	16.964	20.536
24	7.784	9.623	12.221	14.556	17.554	21.429
25	7.843	9.743	12.459	14.929	18.133	22.321
26	7.896	9.853	12.684	15.288	18.702	23.214
27	7.943	9.954	12.898	15.635	19.261	24.107
28	7.984	10.047	13.100	15.970	19.810	25.00
29	8.022	10.132	13.291	16.292	20.349	25.893
30	8.055	10.211	13.472	16.603	20.879	26.786

$$E_1 = \alpha + [(1 - t)p + h]P/X(d,g,n) + (1 - \alpha)[(1 - t) \frac{P/X(d,0,m)}{P/X(i,0,m)} + (t) \frac{P/X(d,i,m)}{P/X(0,i,m)}] \quad (12-10)$$

where

α is the downpayment rate in the terms of the loan, and fixed mortgage payment is assumed,

t is the effective income tax rate of the owner,

p is the property tax rate based on initial capital cost (first year market value),

h is the insurance premium rate,

g is the inflation rate for general cost of goods and services (general inflation rate),

i is the interest rate of the loan,

m is the term (years) of the loan.

Values of $P/X(a,b,c)$ may be determined from Tables 12-1 through 12-6 by referring to the appropriate values in the tables as indicated by the terms in the parentheses following P/X . For example, $P/X(d,0,m)$ may be determined by consulting the appropriate discount rate d , rate of annual increase 0, (zero), and years m .

EXAMPLE 12-1

Determine the economic factors for costs of operation, maintenance, and fuel, E_o , E_m , and E_f , if the annual rate of increase for operating cost, r_o , is 10 percent, annual rate of increase for maintenance, r_m , is 6 percent, annual rate of increase for fuel, r_f , is 12 percent, and the discount rate is 8 percent for a life span of 20 years.

Solution:

$$E_o = P/X(8,10,20) = 22.169 \text{ (from Table 12-4)}$$

$$E_m = P/X(8,6,20) = 15.596 \text{ (from Table 12-4)}$$

$$E_f = P/X(8,12,20) = 26.740 \text{ (from Table 12-4)}$$

EXAMPLE 12-2

Determine the economic factor for interest, insurance, taxes, and other costs, E_1 , if the terms of the loan are $m = 25$ years, $i = 10$ percent, and $\alpha = 20$ percent downpayment. The property tax rate, p , is 3 percent and insurance rate, h , is 0.3 percent of market value, general inflation, g , is 6 percent, and the effective income tax rate is 35 percent. The market discount rate, d , is 8 percent.

Solution:

For Equation (12-10), find appropriate P/X values from the tables.

$$P/X (d,g,n) = P/X (8,6,20) = 15.596$$

$$P/X (d,0,m) = P/X (8,0,25) = 10.675$$

$$P/X (i,0,m) = P/X (10,0,25) = 9.077$$

$$P/X (d,i,m) = P/X (8,10,35) = 29.103$$

$$P/X (0,i,m) = P/X (0,10,25) = 98.347$$

$$\alpha = 0.20$$

$$t = 0.35$$

$$p = 0.03$$

$$h = 0.003$$

Thus,

$$\begin{aligned} E_1 = & 0.20 + [(1-.35)(0.03)+0.003](15.596) \\ & + (1-0.2)[(1-.35) \frac{10.675}{9.077} + (0.35) \frac{29.103}{98.347}] \end{aligned}$$

$$E_1 = 1.245 \quad (\text{ans})$$

EXAMPLE 12-3

Determine the present values of life-cycle costs of a solar system, a non-solar system and the savings with a solar system, given the following information:

$A = 500 \text{ ft}^2$	$r_m = 6\%$
$C_a = 26 \text{ \$/ft}^2$	$r_f = 12\%$
$C_o = 87 \text{ \$/yr}$	$m = 25 \text{ years}$
$C_{oc} = 20 \text{ \$/yr}$	$i = 10\%$
$C_m = 100 \text{ \$/yr}$	$a = 20\% \text{ down}$
$C_{mc} = 10 \text{ \$/yr}$	$p = 3\% \text{ of market value}$
$F = 0.68$	$h = 0.3\% \text{ of market value}$
$c_f = 10.25 \text{ R/MMBtu}$	$g = 6\%$
$L = 130 \text{ MMBtu}$	$t = 35\%$
$r_o = 10\%$	$d = 8\%$

Solution:

The equation to apply for the solar system is Equation (12-13).

From Example 12-1, $E_o = 22.169$, $E_m = 15.596$, $E_f = 26.740$.

From Example 12-2, $E_1 = 1.245$

Therefore,

$$C_T = (500)(26)(1.245) + (87)(22.169) + (100)(15.596) \\ + (1 - .68)(130)(10.25)(26.740)$$

$$C_T = \$31,075 \text{ present value (total cost) over 20 years} \\ \text{of life}$$

The equation to apply to the non-solar system is Equation (12-4).

$$C_{TC} = (20)(22.169) + (20)(15.596) + (130)(10.25)(26.740)$$

$$C_{TC} = \$36,230 \text{ present value (total cost) over 20 years} \\ \text{of life}$$

The cost of the non-solar is clearly larger than the cost of the solar system. The difference, or savings realizable with the solar system, is:

$$\begin{array}{l} \text{Present value} \\ \text{of savings} \end{array} = C_{TC} - C_T = 36,230 - 31,075 = \$5155$$

While in Example 12-3 the present values of the total costs for systems and life time savings are determinable, the calculations are restricted to fixed annual increases, fixed discount rates, fixed property tax and insurance rates, and fixed income tax rates for the owner. These rates are, of course, uncertain in future years and highly variable. If variable rates are to be applied, a detailed year-by-year analysis of cash flow and present worth discounting must be carried out, using the basic form of Equations (12-1) and (12-2).

Annual cash flows are calculated for a system and the annual cost may be discounted to present value. The cost in a future year may be discounted to present worth by multiplying the cost by the present worth factor, P , in:

$$P = \frac{1}{(1+d)^q} \quad (12-11)$$

where

q is any year in the analysis period from 1 to n

d is the market discount rate

Values of P for practical ranges of d and q are tabulated in Table 12-7.

ENERGY COSTS

The conversion of unit costs of energy to dollars per million Btu (\$/MMBtu) with various furnace efficiencies is shown in Figure 12-1 for natural gas, propane, and No. 2 fuel oil. The conversion of electrical energy costs to dollars per million Btu for resistance heating and heat

Table 12-7
 Present Worth Factors (P)
 (use for Worksheet LCA-4)

Year of Analysis	Discount Rate										
	6	7	8	9	10	11	12	13	14	15	16
1	.943	.935	.926	.917	.909	.901	.893	.885	.877	.870	.862
2	.890	.873	.857	.842	.826	.812	.797	.783	.769	.756	.743
3	.840	.816	.794	.772	.751	.731	.712	.693	.675	.658	.641
4	.792	.763	.735	.708	.683	.659	.636	.613	.592	.572	.552
5	.747	.713	.681	.650	.621	.593	.567	.543	.519	.497	.476
6	.705	.666	.630	.596	.564	.535	.507	.480	.456	.432	.410
7	.665	.623	.583	.547	.513	.482	.452	.425	.400	.376	.354
8	.627	.582	.540	.502	.467	.434	.404	.376	.351	.327	.305
9	.592	.544	.500	.460	.424	.391	.361	.333	.308	.284	.263
10	.558	.508	.463	.422	.386	.352	.322	.295	.270	.247	.227
11	.527	.475	.429	.388	.350	.317	.287	.261	.237	.215	.195
12	.497	.444	.397	.356	.319	.286	.257	.231	.208	.187	.168
13	.469	.415	.368	.326	.290	.258	.229	.204	.182	.163	.145
14	.442	.388	.340	.299	.263	.232	.205	.181	.160	.141	.125
15	.417	.362	.315	.275	.239	.209	.183	.160	.140	.123	.108
16	.394	.339	.292	.252	.218	.188	.163	.141	.123	.107	.093
17	.371	.317	.270	.231	.198	.170	.146	.125	.108	.093	.080
18	.350	.296	.250	.212	.180	.153	.130	.111	.095	.081	.069
19	.331	.277	.232	.194	.164	.138	.116	.098	.083	.070	.060
20	.312	.258	.215	.178	.149	.124	.104	.087	.073	.061	.051
21	.294	.242	.199	.164	.135	.112	.093	.077	.064	.053	.044
22	.278	.226	.184	.150	.123	.101	.083	.068	.056	.046	.038
23	.262	.211	.170	.138	.112	.091	.074	.060	.049	.040	.033
24	.247	.197	.158	.126	.102	.082	.066	.053	.043	.035	.028
25	.233	.184	.146	.116	.092	.074	.059	.047	.038	.030	.024

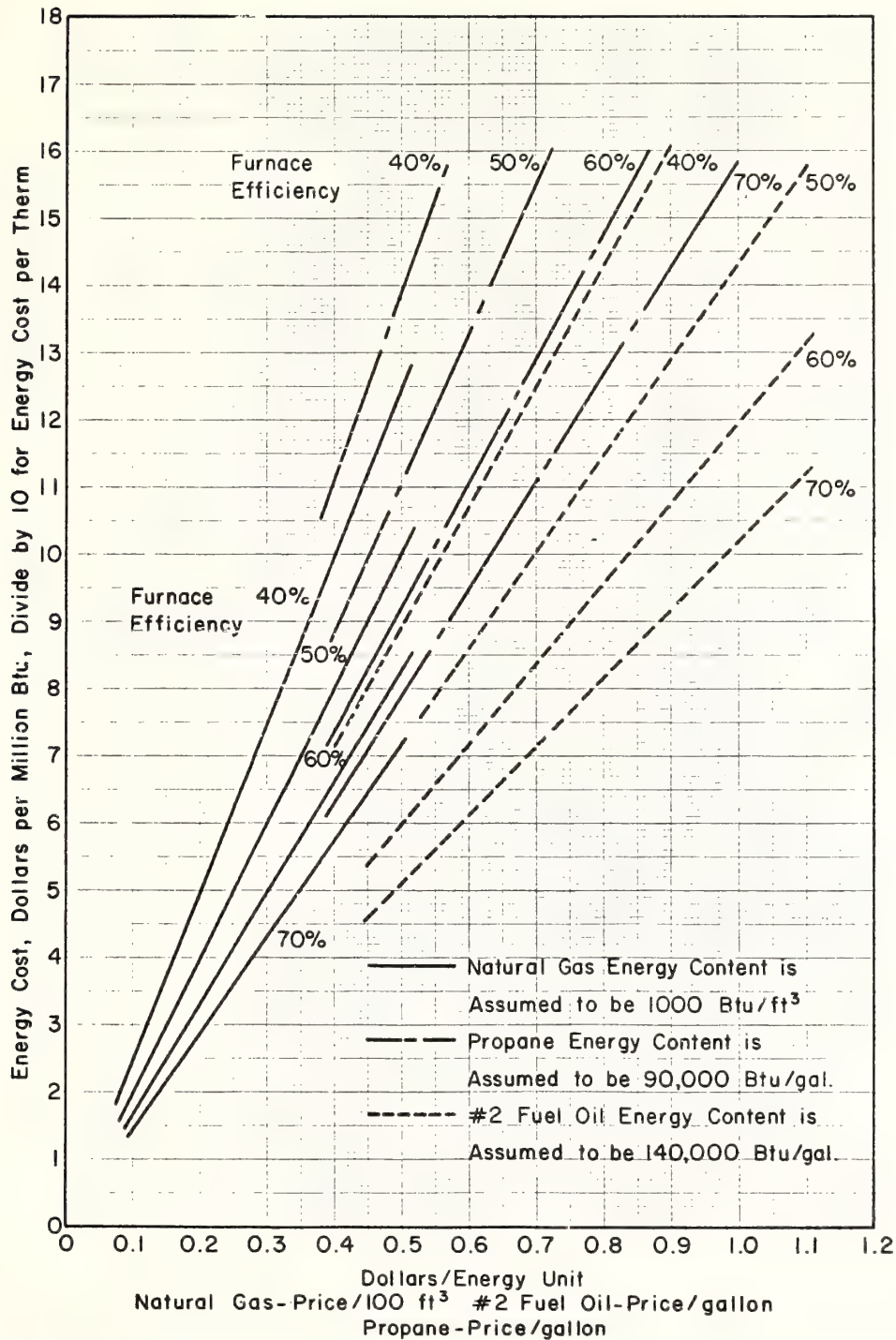


Figure 12-1. Energy Cost per Million Btu for Natural Gas, Propane, and No. 2 Fuel Oil

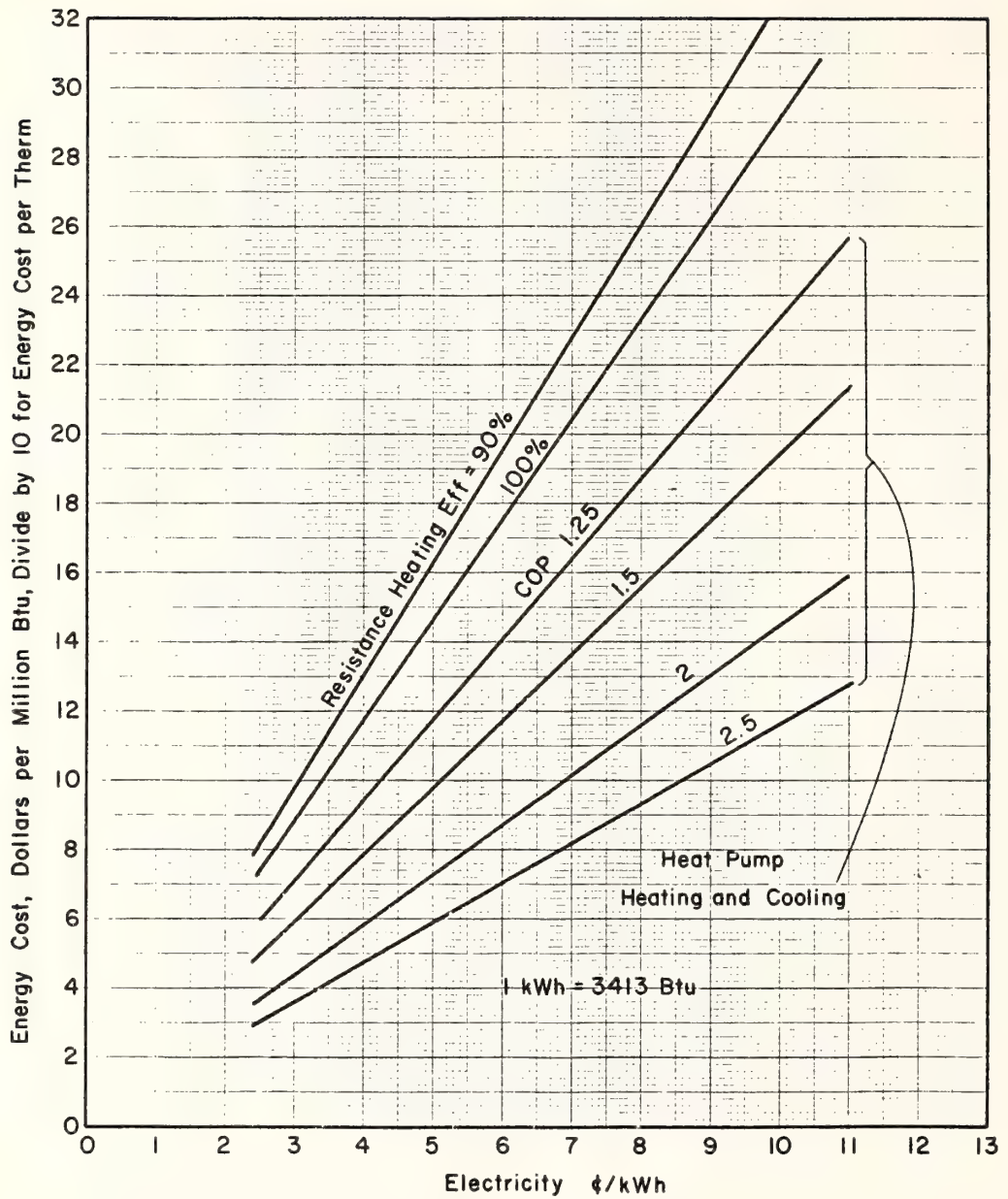


Figure 12-2. Energy Cost per Million Btu for Electricity

pumps with various coefficients of performance is shown in Figure 12-2. To determine the cost per million Btu of heat generated from furnaces, electric resistance heaters, or heat pumps, follow the unit cost of energy, found on the horizontal axis of the graphs, vertically to the appropriate line on the graph and read the cost in dollars along the vertical axis. For example, if No. 2 fuel oil cost is one dollar per gallon, and the furnace efficiency is 60 percent, the energy cost is \$12.00/MMBtu or \$1.20 per therm. If the furnace is more efficient, say 70 percent, the energy cost is \$10.29 MMBtu. Similarly, if electricity cost is five cents per kilowatt-hour ($\$/\text{kWh}$), and resistance heating is used at 100 percent efficiency, the energy cost is \$14.65/MMBtu. If a heat pump is used, and the COP of the heat pump is 2, the energy cost is \$7.32/MMBtu.

The cost of energy will increase in future years, and an estimate of the rate of increase is subject not only to inflation rates of goods and services, but also to economic and political decisions of the Federal Government and the governments of other nations. One expects, however, the rate of fuel cost increases to be different from general inflation rates and higher by a few percent, at least for the immediate future.

INFLATION RATES

The increases in costs per unit of energy several years in the future in terms of cents per gallon of oil, cents per kilowatt-hour of electricity, cents per hundred cubic feet of natural gas, or dollars per therm, can be estimated on the basis of annual percentage increases over current costs. The multiplying factors for current energy costs to

determine future costs are shown in Figure 12-3. The horizontal axis is the number of years beyond the current year. The vertical axis is the multiplying factor over current costs, and is simply the interest compounded annually, $(1 + i)^n$, where i is the interest rate and n is years.

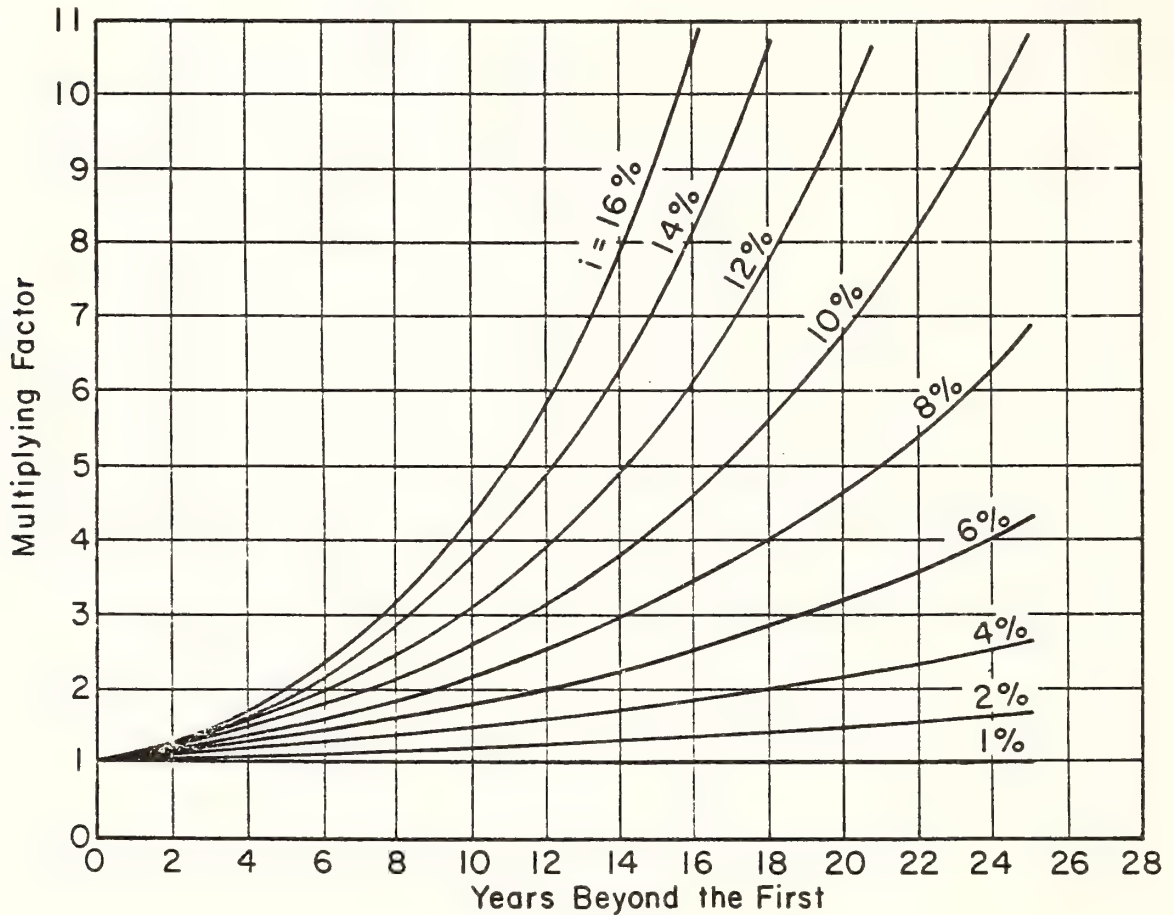


Figure 12-3. Inflation Factors

For example, if the current cost of electricity is expected to increase at a rate of 6 percent each year for the next 12 years, Figure 12-3 shows that at the end of 12 years, the electricity cost will double. If the current electricity cost is 5 cents per kilowatt-hour, equivalent to \$14.65 per million Btu, the cost will be 10 cents per kilowatt-hour and \$29.30 per million Btu in 12 years.

SOLAR SYSTEM COSTS

There is considerable uncertainty about the installed costs of solar heating systems, and there is insufficient information available to substantiate published reports. System costs based on research projects and demonstration projects funded by the Federal Government are misleading, because the total costs of such projects may include considerable design and engineering. In some instances, instrumentation for monitoring the performance of experimental systems and the cost of developing alternative components are included. The costs reported in popular magazines and newspaper accounts are likewise misleading, because they are often based on systems which have been designed and assembled by the owner on a do-it-yourself basis. Installation labor, normally a major cost item, is seldom included.

Guidelines are provided in this section to estimate the total installed cost of a solar system including equipment and installation labor. Of these two items, equipment costs are the larger and easier to estimate, largely by consulting manufacturer's literature and price lists. Estimating labor costs is more difficult because it depends upon the type of installation, location of the house, experience of the installer, and other factors. Ranges in equipment price and estimates of man-hours for installation of systems in new buildings are listed below to provide cost-estimating guidelines. There may be specific items of equipment which have lower costs than those listed, and some which are more expensive. The prices may not be representative of any single project.

EQUIPMENT AND INSTALLATION TIME ESTIMATESLiquid-Heating Systems

Table 12-8

Typical Equipment and Material Prices (in 1978)
for Liquid-Heating Systems

Item	Unit	Price Range (in dollars)		
		Low	Medium	High
Flat-plate collectors and mounting hardware	ft ²	10	15	24
Storage tank	750-1200 gal capacity	1000	1500	2500
Pumps and motor	10-20 gpm	80	180	350
Heat exchanger	each	200	300	400
Controls and sensors	each	500	750	1500
Piping (3/4-inch copper)	ft	.45	.60	.80-.85
Valves	each	20	30	45
Misc. fittings	-	200	250	350
Expansion tank	-	60	80	100
Insulation	-	500	750	1000
DHW Preheat tank	each	80	100	150

Table 12-9
Installation Time Estimates for Typical Liquid-Heating Systems

Item	Unit	Time (man-hours)		
		Low	Medium	High
Collectors and flashing	400-500 ft ²	40	60	80
Storage tank	each	8	10	12
Piping loops	all	40	60	80
DHW preheat subsystem	-	8	12	20
Insulation	all	16	20	30
Controls	-	8	12	16
Testing and balancing	-	10	15	20

Air-Heating Systems

Table 12-10
Typical Equipment and Material Prices (in 1978)
for Air-Heating Systems

Item	Unit	Price Range (in dollars)		
		Low	Medium	High
Flat-plate collectors and mounting hardware	ft ²	10	15	24
Storage containers	ft ³	0.5	1	1.5
Gravel	ton	3	4	5
Blower and motor	each	150	175	200
Control and sensors	set	500	750	1500
Motorized dampers	each	115	125	150
Heat exchanger	each	45	60	80
DHW Preheat tank	each	80	100	150
Ducts	bulk	2000	2500	3500
Insulation	bulk	500	750	1000
Miscellaneous	-	200	300	400

Table 12-11

Installation Time Estimates for Typical Air-Heating Systems

Item	Unit	Time (man-hours)		
		Low	Medium	High
Collectors	400-500 ft ²	40	60	80
Storage unit	each	20	25	30
Ducting	all	50	75	100
Controls	-	8	12	16
DHW preheat subsystem	-	8	12	20
Insulation	-	16	20	30
Testing and balancing	-	10	15	20

Domestic Water Heaters

There are numerous manufacturers marketing complete systems for domestic water heating. Prices of "packaged units" vary from about \$1500 to about \$4000 with a median price of about \$2500. Solar water heaters may also be assembled from components supplied from different manufacturers. Price ranges of components are listed in Table 12-12.

Installed cost of a "packaged" solar domestic hot water unit appears to be approximately the same as the installed cost of a system assembled with individually selected components. Manufacturers of packaged systems supply an integrated set of components which mainly saves time in designing and purchasing the components separately. Because components are pre-arranged to fit together, some savings in assembly time can be expected. Time required for installation of solar hot water systems will depend on background experience of the installer, and estimates are provided in Table 12-13.

Table 12-12

Equipment Costs for Components of Domestic Water Heaters (1978 prices)

Item	Unit	Price Range (in dollars)		
		Low	Medium	High
Flat-plate collectors and mounting hardware	ft ²	10	15	24
Preheat tank	80-gallon capacity	100	200	250
Pump and motor assembly	each	80	120	150
Heat exchanger	each	200	300	400
Controls and sensors	set	100	150	200
Piping (1/2-inch copper)	ft	.40	.50	.75
Valves	each	20	30	45
Miscellaneous fittings	bulk	30	40	50
Expansion tank	each	60	80	100
Insulation	bulk	80	100	120

Table 12-13

Estimates of Installation Times for Domestic Water Heaters

Item	Unit	Time (man-hours)		
		Low	Medium	High
Collectors and flashing	2-4 collector units	4	8	12
Preheater tank	1	1	2	2
Piping loops, pumps, valves	all	8	12	16
Insulation	-	2	4	6
Controls	-	1	2	3
Filling, testing, and adjusting	-	2	4	6
For packaged units, subtract 20 percent from total	Total (with 20% subtracted)	14	26	37

TYPICAL INSTALLED COSTSSpace Heating System - Liquid Collectors

An estimate for the installed cost of a typical liquid-heating system in a new building with 400 ft² of collectors is outlined below using the median values in Tables 12-8 and 12-9.

1.	Collectors	equipment 400 ft ² x \$15/ft ²	\$6,000
		installation 60 hrs x \$15/hr	900
2.	Storage Tank	equipment	1,500
		installation 10 hrs x \$15/hr	150
3.	Pipe Loops	equipment	2,070
		installation 60 hrs x \$15/hr	900
4.	DHW Subsystem	equipment	280
		installation 12 hrs x \$15/hr	180
5.	Controls	equipment	750
		installation 12 hrs x \$15/hr	180
6.	Insulation	materials	750
		installation 20 hrs x \$15/hr	300
7.	Testing and balancing		<u>225</u>
Total Estimated Costs			<u>14,185</u>
Breakdown of costs:			
Equipment & materials			11,350
Labor			<u>2,835</u>
Installed cost/unit collector area			\$35.46/ft ²

Space Heating System - Air Collectors

An estimate of the installed cost of a typical air-heating system in a new building with 400 ft² of collectors is outlined below using the median values in Tables 12-10 and 12-11.

1.	Collectors	equipment	400 ft ² X \$15/ft ²	\$6,000
		installation	60 hrs x \$15/hr	900
2.	Pebble Bed	Container	300 ft ³ x \$1/ft ³	300
		Gravel	15 tons x \$4/ton	60
		Assembly	25 hrs x \$15/hr	375
3.	Duct, Pumpers & blowers	equipment		3,175
		installation	75 x \$15/hr	1,125
4.	DHW Subsystem	equipment		385
		installation	12 x \$15/hr	180
5.	Controls	equipment		750
		installation	12 hrs x \$15/hr	180
6.	Insulation	materials		750
		installation	20 hrs x \$15/hr	300
7.	Testing and balancing		15 hrs x \$15/hr	<u>225</u>
Total Estimated Costs				14,705
Breakdown of costs:				
Equipment & materials				11,420
Labor				<u>3,285</u>
Installed cost/unit collector area				\$36.76/ft ²

Domestic Water Heaters

Two estimates for the installed cost of a typical solar domestic hot water system in a new building with 60 ft² of collectors are provided below. One estimate is for the installed cost of a "packaged" unit, and the second is for assembling a system from separately purchased components.

1.	Complete equipment (3 modules = 60 ft ²)	\$2500
	Complete installation 26 hrs x \$15/hr	<u>320</u>
Total Estimated Cost		<u>\$2890</u>
Installed cost/unit collector area		<u><u>\$48.17/ft²</u></u>

1.	Collectors and flashing	equipment (3 modules) 60 ft ² x \$15/ft ²	\$ 900
		installation 8 hrs x \$15/hr	120
2.	Preheater tank	equipment	200
		installation 2 hrs x \$15/hr	30
3.	Pumps and motor	equipment (2)	240
		installation (included in piping)	
4.	Heat exchanger	equipment	300
		installation (included in piping)	
5.	Piping	materials	340
		installation 12 hrs x \$15/hr	180
6.	Insulation	materials	100
		installation 4 hrs x \$15/hr	60
7.	Controls and sensors	equipment	150
		installation 2 hrs x \$15/hr	30
8.	Filling, testing and adjusting	4 hrs x \$15/hr	<u>60</u>

Total Estimated Costs \$2,710

Breakdown of costs:

Equipment and materials 2,230

Installation 480

Installed cost/unit collector area \$45.17/ft²

MORTGAGE PAYMENTS

The largest portion of the annual cost of a solar system is the repayment of the loan obtained to install the system. The loan may be based on the total building costs or separately on the solar system alone. In either event, a downpayment of up to 30 percent may be required to obtain the loan.

The annual mortgage payments can be calculated from the mortgage interest rate and term of the loan using the curves of Figure 12-4. To illustrate the use of Figure 12-4, suppose that a solar system with 400 square feet of collectors costs \$14,185. A 25-year loan is obtained to purchase and install the system with interest at 10 percent, which requires a 20 percent downpayment. The annual mortgage payment on the loan is calculated as follows:

$$\begin{array}{l} \text{Annual} \\ \text{Mortgage} = (\text{System cost} - \text{downpayment}) \times (\text{Annual repayment factor}) \\ \hspace{15em} \text{(from Figure 12-4)} \end{array}$$

or

$$\begin{array}{l} \$ \quad 1248 = (14,185 - 2837) \times (0.11) \end{array}$$

TAX CREDIT

A major item of income tax credit is the legislation passed by the Federal Government, effective January 1980 through 1985, which allows a maximum credit of \$4000 from the owner's Federal income tax liability. The credit allowance is 40 percent of the first \$10,000 of the cost of a qualified solar system. In some states there are additional income tax credits or deductions provided to owners of solar systems. A list

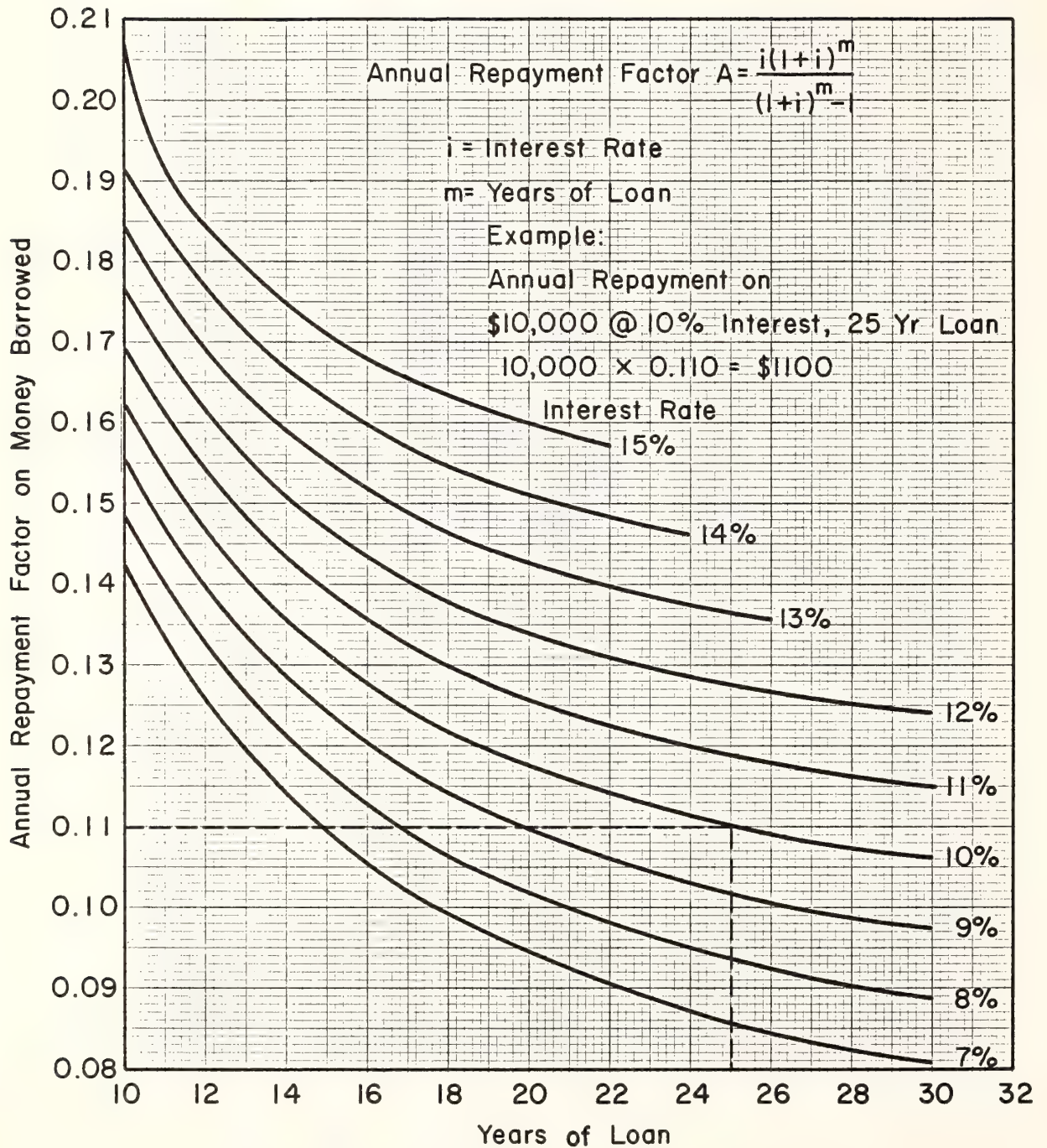


Figure 12-4. Repayment on Loan

of state provisions prepared by the National Solar Heating and Cooling Information Center is appended to this module.

PROPERTY TAXES

Property taxes are usually based on a fraction of the assessed value of the solar system. The method of assessment and the tax rate vary from state to state and sometimes from county to county within a state. The office of the county treasurer can provide detailed information on assessment practices and the tax rate. Usually, the assessed value is a fraction of the market value of the property, and the tax rate is applied to the assessed value. The amount of property tax on the solar system can be calculated as follows:

$$\text{Property tax} = (\text{System cost}) \times (\text{Fraction of assessed value}) \times (\text{tax rate})$$

INSURANCE

Insurance rates on houses with a solar system, at present, are the same as for houses without solar systems. The solar equipment is therefore usually insured at a rate equivalent to that applicable to all the conventional components of the building. The basic insurance rate depends upon the type of house construction and location of the building within or outside a city or town. The insurance rate for a comprehensive homeowners policy differs from the rate for a straight fire insurance policy, and the insurance rates for earthquake and flood damage (which are federally subsidized) are the only special insurances available for owners of buildings. Information on various insurance

rates is available from local insurance agents. Very few insurance companies have established insurance rates for solar systems. Damage to the contents of a building resulting from leaks in piping or storage tanks or damage to the solar system resulting from flooding by natural causes is based on comprehensive or flood insurance rates. Although there are many factors to be considered, the annual premium on insurance for houses with solar systems is less than one percent of the value of the house and contents, and ranges from 0.3 to about 0.6 percent.

ADDITIONAL INCOME TAX CREDITS

The "savings" on state and Federal income taxes for property tax and interest paid on the mortgage can be substantial, depending upon the "tax bracket" of the homeowner. The amount of interest paid annually on the mortgage decreases with the number of years remaining on the mortgage. The portion of annual mortgage which is paid as interest can be determined from Tables 12-14 through 12-18. The use of the tables is illustrated in the following example.

Let us assume that a loan of \$11,348 has been secured with a term of 25 years and 10 percent interest. The annual mortgage payment was computed in the previous section to be \$1248. Of that mortgage payment, 90.8 percent, or \$1133, is for interest in the first year. From Table 12-17, line 1, at the interest rate of 10 percent, the number 0.908 indicates the fraction of the mortgage payment which is interest on the money borrowed and payable in the first year. Mortgages are usually repaid monthly, but for this analysis annual payment figures will

Table 12-14

Fraction of Mortgage Payment (A) Which is Interest

$$I/A = 1 - (1+i)^{j-m-1} \quad \text{Mortgage Term (m) = 10 years}$$

YEAR (j)	INTEREST RATE									
	6	7	8	9	10	11	12	13	14	15
1	.442	.492	.537	.578	.614	.648	.678	.705	.730	.753
2	.408	.456	.500	.540	.576	.609	.639	.667	.692	.716
3	.373	.418	.460	.498	.533	.566	.596	.624	.649	.673
4	.335	.377	.417	.453	.487	.518	.548	.575	.600	.624
5	.295	.334	.370	.404	.436	.465	.493	.520	.544	.568
6	.253	.287	.319	.350	.379	.407	.433	.457	.481	.503
7	.208	.237	.265	.292	.317	.341	.364	.387	.408	.428
8	.160	.184	.206	.228	.249	.269	.288	.307	.325	.342
9	.110	.127	.143	.158	.174	.188	.203	.217	.231	.244
10	.057	.065	.074	.083	.091	.099	.107	.115	.123	.130

Table 12-15

Fraction of Mortgage Payment (A) Which is Interest
 $I/A = 1 - (1+i)^{j-m-1}$ Mortgage Term (m) = 15 years

YEAR (j)	INTEREST RATE									
	6	7	8	9	10	11	12	13	14	15
1	.583	.638	.685	.725	.761	.791	.817	.840	.860	.877
2	.588	.612	.660	.701	.737	.768	.795	.819	.840	.859
3	.531	.585	.632	.674	.710	.742	.771	.796	.818	.837
4	.503	.556	.603	.644	.681	.714	.743	.769	.792	.813
5	.473	.525	.571	.612	.650	.683	.713	.739	.763	.785
6	.442	.492	.537	.578	.614	.648	.678	.705	.730	.753
7	.408	.456	.500	.540	.576	.609	.639	.667	.692	.716
8	.373	.418	.460	.498	.533	.566	.596	.624	.649	.673
9	.335	.377	.417	.453	.487	.518	.548	.575	.600	.624
10	.295	.334	.370	.404	.436	.465	.493	.520	.544	.568
11	.253	.287	.319	.350	.379	.407	.433	.457	.481	.503
12	.208	.237	.265	.292	.317	.341	.364	.387	.408	.428
13	.160	.184	.206	.228	.249	.269	.288	.307	.325	.342
14	.110	.127	.143	.158	.174	.188	.203	.217	.231	.244
15	.057	.065	.074	.083	.091	.099	.107	.115	.123	.130

Table 12-16

Fraction of Mortgage Payment (A) Which is Interest

$$I/A = 1 - (1+i)^{j-m-1} \quad \text{Mortgage Term (m) = 20 years}$$

YEAR (j)	INTEREST RATE									
	6	7	8	9	10	11	12	13	14	15
1	.688	.742	.785	.822	.851	.876	.896	.913	.927	.939
2	.669	.723	.768	.806	.836	.862	.884	.902	.917	.930
3	.650	.704	.750	.788	.820	.847	.870	.889	.905	.919
4	.629	.683	.730	.769	.802	.830	.854	.875	.892	.907
5	.606	.661	.708	.748	.782	.812	.837	.859	.877	.893
6	.583	.638	.685	.725	.761	.791	.817	.840	.860	.877
7	.558	.612	.660	.701	.737	.768	.795	.819	.840	.859
8	.531	.585	.632	.674	.710	.742	.771	.796	.818	.837
9	.503	.556	.603	.644	.681	.714	.743	.769	.792	.813
10	.473	.525	.571	.612	.650	.683	.713	.739	.763	.785
11	.442	.492	.537	.578	.614	.648	.678	.705	.730	.753
12	.408	.456	.500	.540	.576	.609	.639	.667	.692	.716
13	.373	.418	.460	.498	.533	.566	.596	.624	.649	.673
14	.335	.377	.417	.453	.487	.518	.548	.575	.600	.624
15	.295	.334	.370	.404	.436	.465	.493	.520	.544	.568
16	.253	.287	.319	.350	.379	.407	.433	.457	.481	.503
17	.208	.237	.265	.292	.317	.341	.364	.387	.408	.428
18	.160	.184	.206	.228	.249	.269	.288	.307	.325	.342
19	.110	.127	.143	.158	.174	.188	.203	.217	.231	.244
20	.057	.065	.074	.083	.091	.099	.107	.115	.123	.130

Table 12-17

Fraction of Mortgage Payment (A) Which is Interest

 $I/A = 1 - (1+i)^{j-m-1}$ Mortgage Term (m) = 25 years

YEAR (j)	INTEREST RATE									
	6	7	8	9	10	11	12	13	14	15
1	.767	.816	.854	.884	.908	.926	.941	.953	.962	.970
2	.753	.803	.842	.874	.898	.918	.934	.947	.957	.965
3	.738	.789	.830	.862	.888	.909	.926	.940	.951	.960
4	.722	.774	.816	.850	.877	.899	.917	.932	.944	.954
5	.706	.758	.801	.836	.865	.888	.907	.923	.936	.947
6	.688	.742	.785	.822	.851	.876	.896	.913	.927	.939
7	.669	.723	.768	.806	.836	.862	.884	.902	.917	.930
8	.650	.704	.750	.788	.820	.847	.870	.889	.905	.919
9	.629	.683	.730	.769	.802	.830	.854	.875	.892	.907
10	.606	.661	.708	.748	.782	.812	.837	.859	.877	.893
11	.583	.638	.685	.725	.761	.791	.817	.840	.860	.877
12	.558	.612	.660	.701	.737	.768	.795	.819	.840	.859
13	.531	.585	.632	.674	.710	.742	.771	.796	.818	.837
14	.503	.556	.603	.644	.681	.714	.743	.769	.792	.813
15	.473	.525	.571	.612	.650	.683	.713	.739	.763	.785
16	.442	.492	.537	.578	.614	.648	.678	.705	.730	.753
17	.408	.456	.500	.540	.576	.609	.639	.667	.692	.716
18	.373	.418	.460	.498	.533	.566	.596	.624	.649	.673
19	.335	.377	.417	.453	.487	.518	.548	.575	.600	.624
20	.295	.334	.370	.404	.436	.465	.493	.520	.544	.568
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22	.208	.237	.265	.292	.317	.341	.364	.387	.408	.428
23	.160	.184	.206	.228	.249	.269	.288	.307	.325	.342
24	.110	.127	.143	.158	.174	.188	.203	.217	.231	.244
25	.057	.065	.074	.083	.091	.099	.107	.115	.123	.130

Table 12-18

Fraction of Mortgage Payment (A) Which is Interest

 $I/A = 1 - (1+i)^{j-m-1}$ Mortgage Term (m) = 30 years

YEAR (j)	INTEREST RATE									
	6	7	8	9	10	11	12	13	14	15
1	.826	.869	.901	.925	.943	.956	.967	.974	.980	.985
2	.815	.859	.893	.918	.937	.952	.963	.971	.978	.983
3	.804	.850	.884	.910	.931	.946	.958	.967	.974	.980
4	.793	.839	.875	.902	.924	.940	.953	.963	.971	.977
5	.780	.828	.865	.894	.916	.934	.947	.958	.967	.974
6	.767	.816	.854	.884	.908	.926	.941	.953	.962	.970
7	.753	.803	.842	.874	.898	.918	.934	.947	.975	.965
8	.738	.789	.830	.862	.888	.909	.926	.940	.951	.960
9	.722	.774	.816	.850	.877	.899	.917	.932	.944	.954
10	.706	.758	.801	.836	.865	.888	.907	.923	.936	.947
11	.688	.742	.785	.822	.851	.876	.896	.913	.927	.939
12	.669	.723	.768	.806	.836	.862	.884	.902	.917	.930
13	.650	.704	.750	.788	.820	.847	.870	.889	.905	.919
14	.629	.683	.730	.769	.802	.830	.854	.875	.892	.907
15	.606	.661	.708	.748	.782	.812	.837	.859	.877	.893
16	.583	.638	.685	.725	.761	.791	.817	.840	.860	.877
17	.558	.612	.660	.701	.737	.768	.795	.819	.840	.859
18	.531	.585	.632	.674	.710	.742	.771	.796	.818	.837
19	.503	.556	.603	.644	.681	.714	.743	.769	.792	.813
20	.473	.525	.571	.612	.650	.683	.713	.739	.763	.785
21	.442	.492	.537	.578	.614	.648	.678	.705	.730	.753
22	.408	.456	.500	.540	.576	.609	.639	.667	.692	.716
23	.373	.418	.460	.498	.533	.566	.596	.624	.649	.673
24	.335	.377	.417	.453	.487	.518	.548	.575	.600	.624
25	.295	.334	.370	.404	.436	.465	.493	.520	.544	.568
26	.253	.287	.319	.350	.379	.407	.433	.457	.481	.503
27	.208	.237	.265	.292	.317	.341	.364	.387	.408	.428
28	.160	.184	.206	.228	.249	.269	.288	.307	.325	.342
29	.110	.127	.143	.158	.174	.188	.203	.217	.231	.244
30	.057	.065	.074	.083	.091	.099	.107	.115	.123	.130

suffice. In the eleventh year, the interest paid during the year is $(0.791) \times (\$1248)$, or \$987. The income tax savings on a Federal or state return for interest and taxes would be:

$$\left(\frac{\text{Income}}{\text{tax credit}} \right) = \left(\frac{\text{Interest and}}{\text{taxes}} \right) \times \left(\frac{\text{Tax rate based}}{\text{on net income}} \right) .$$

The Federal income tax return provides credit for state income taxes paid and many states allow credit for Federal income taxes. Thus the full credit for tax savings resulting from payment of interest is not simply the sum of state and Federal tax savings. The net effective rate for states allowing credit is:

$$\left(\frac{\text{Net}}{\text{Effective}} \right) \left(\frac{\text{Rate}}{\text{Rate}} \right) = \left(\frac{\text{Federal}}{\text{tax rate}} \right) + \left(\frac{\text{State}}{\text{tax rate}} \right) - 2 \left(\frac{\text{Federal}}{\text{tax rate}} \right) \times \left(\frac{\text{State}}{\text{tax rate}} \right)$$

For states which do not give credit, the net effective rate is:

$$\left(\frac{\text{Net}}{\text{Effective}} \right) \left(\frac{\text{Rate}}{\text{Rate}} \right) = \left(\frac{\text{Federal}}{\text{tax rate}} \right) + \left(\frac{\text{State}}{\text{tax rate}} \right) - 1 \left(\frac{\text{Federal}}{\text{tax rate}} \right) \times \left(\frac{\text{State}}{\text{tax rate}} \right)$$

If the income tax rate on a Federal tax return is 25 percent and on a state tax return is 10 percent, the net effective rate is $(0.25 + 0.10 - 2 \times 0.25 \times 0.10) = 0.30$, or 30 percent. The net annual income tax savings realized on the Federal and state taxes for interest paid is $(0.30) \times (\$1133)$, or \$340 in the first year and $(0.30) \times (\$987)$ or \$296 in the eleventh year.

OPERATING COSTS

The cost of operating a solar heating system, including the cost for operating the auxiliary unit in the system, is the cost of electric energy required to operate the pumps, central heat distribution fan, valves, and controller in a hydronic system, and the blowers, motorized

dampers, and controller in an air system. The amount of energy used to collect, store, and distribute solar energy varies from system to system in the range from 5 to 10 percent of the total solar energy collected. The lower values in the range apply to low-head systems with small pressure drops, and air systems with single blowers with small pressure drops. The higher values in the range apply to high-head systems with large pressure drops, small systems with large pumps, and air systems with two blowers.

The operating cost of a non-solar system is considerably less than that for a solar system. Although the blower size for distributing air to the rooms is the same, the power requirement is less for a non-solar system because the pressure drop in the system is lower. As an approximation, the energy required to operate a non-solar system is two to three percent of the total annual heating load.

MAINTENANCE COSTS

The maintenance costs for solar systems are unknown because there is insufficient long-term experience with various systems to indicate an appropriate maintenance cost. While there is one air system that has operated continuously for over 20 years, on which the maintenance cost has been negligible, it can be expected that all solar systems should receive annual preventive maintenance requiring about one man-day per year. Liquid systems may require somewhat more frequent maintenance than once a year. For the purpose of economic analysis, maintenance costs can be estimated at about \$100 for the first year and escalated annually at a selected inflation rate.

ECONOMIC ANALYSIS WORKSHEETS

Included in this section are "short" forms and "long" forms for calculating life-cycle costs of solar and conventional heating systems. The short form enables direct calculation of the present worth of cumulative costs over a predetermined life of the system, e.g. 20 years. The long form enables year-by-year calculations and allows flexibility in the analysis because annual inflation rates can be changed for any item. In using the short form the inflation rates remain constant for the total period of analysis. Calculation procedures are explained through detailed worksheets in the following sections.

WORKSHEET LCA-1

Worksheets LCA-1 (2 sheets) are data sheets to facilitate the computations. Technical, economic, and cost data are listed on the worksheets.

WORKSHEET LCA-2

Worksheet LCA-2 outlines step-by-step procedure for calculating life-cycle costs of both the solar and non-solar systems. The economic factors, E values, are determined from Tables 12-1 through 12-6.

DATA SHEET FOR ECONOMIC ANALYSIS

Project _____

Building Data

1. Annual space heating load _____ MMBtu/yr
2. Annual DHW heating load _____ MMBtu/yr
3. Total H and DHW load (add lines 1 & 2) _____ MMBtu/yr

Solar System Data

4. Collector area _____ ft²
5. Fraction of annual heating load
supplied from solar _____ decimal

Energy Prices

6. c_e , current energy cost for electricity
(use Figure 12-2) _____ ¢/kWh _____ \$/MMBtu
7. c_f , c_{fc} , current cost of fuel
(use Figure 12-1 or 12-2) _____ \$/MMBtu

Terms of Loan

8. m , term of the loan for solar system _____ yrs
9. α , downpayment _____ % _____ decimal
10. i , interest rate on loan _____ % _____ decimal

Economic Data

11. C_a , installed cost of solar system per
unit area _____ \$/ft²
12. r_f , estimated auxiliary fuel inflation
rate _____ %
13. r_e , r_o , estimated electric energy
inflation _____ %
14. g , r_m , estimated general inflation
rate _____ %
15. p , property tax rate (based on
market value) _____ decimal
16. h , insurance premium rate _____ decimal
17. Federal income tax rate for owner _____ decimal
18. State income tax rate for owner _____ decimal
19. t , effective income tax rate
{i.e., (line 17) + (line 18)
- [2 x (line 17) x (line 18)]} _____ decimal
20. d , market discount rate _____ decimal

Solar System Cost Items

21. Installed cost (line 4 x line 11) _____ \$
22. Federal tax credit for solar
(40% of first \$10,000 of
system cost) _____ \$
23. Downpayment (line 21 x line 9) _____ \$
24. Amount of loan (line 21 - line 23) _____ \$
25. Annual mortgage payment (multiply line
24 by annual mortgage rate from
Figure 12-4) _____ \$/yr
26. C_f , first year cost of auxiliary heating
(line 3 x (1-line 5) x line 7) _____ \$/yr
27. First year property tax (line 21 x
line 15) _____ \$/yr
28. First year insurance premium
(line 21 x line 16) _____ \$/yr
29. C_o , first year cost of operating the
solar system (line 3 x (a value
between .05 and .10) x line 6) _____ \$/yr
30. C_m , first year maintenance cost
(estimate) _____ \$/yr

Non-Solar System Cost Items

31. C_{fc} , first year cost of fuel for non-
solar system (line 3 x line 7) _____ \$/yr
32. C_{oc} , first year cost of operating
non-solar system (line 3 x
.01 x line 6) _____ \$/yr

LIFE-CYCLE COST ANALYSIS

Total Cost for Solar System

33. n, total years of analysis _____ yrs
34. A, collector area (line 4 of LCA-1) _____ ft²
35. L, annual heat load (line 3 of LCA-1) _____ MMBtu
36. F, fraction of annual heat provided
by the solar system (line 5 of
LCA-1) _____ decimal
37. P/X (d,g,n) (See Tables 12-1
through 12-6) _____
38. P/X (d,0,m) (See Tables 12-1
through 12-6) _____
39. P/X (i,0,m) (See Tables 12-1
through 12-6) _____
40. P/X (d,i,m) (See Tables 12-1
through 12-6) _____
41. P/X (0,i,m) (see Tables 12-1
through 12-6) _____
42. $(t) \left[\frac{P/X (d,i,m)}{P/X (0,i,m)} \right] = \left(\frac{\text{line 19} \times \text{line 40}}{\text{line 41}} \right)$ _____
43. $(1 - t) \left[\frac{P/X (d,0,m)}{P/X (i,0,m)} \right] = \left[\frac{(1 - \text{line 19}) \times (\text{line 38})}{\text{line 41}} \right]$ _____
44. Add line 42 and line 43 _____
45. $1 - a (1 - \text{line 9})$ _____
46. Multiply: line 44 x line 45 _____
47. $(1-t)(p) + h$
 $(1 - \text{line 19})(\text{line 15}) + (\text{line 16})$ _____
48. Multiply: line 47 x line 37 _____
49. $E_1 = (\text{line 9}) + (\text{line 48}) + (\text{line 46})$ _____
50. $E_0 = P/X (d,r_0,n)$ (see Tables
12-1 through 12-6) _____
51. $E_m = P/X (d,r_m,n)$ (see Tables 12-1
through 12-6) _____

52. $E_f = P/X (d, r_f, n)$ (See Tables 12-1 through 12-6) _____
53. $(A)(C_a)(E_1) = (\text{line 34} \times \text{line 11} \times \text{line 49})$ _____ \$
54. $C_o E_o = (\text{line 29} \times \text{line 50})$ _____ \$
55. $C_m E_m = (\text{line 30} \times \text{line 51})$ _____ \$
56. $(1-F)(L)(c_f)(E_f) = (\text{____})(\text{____})(\text{____})(\text{____})$ _____ \$
(1 - line 36) \times line 35 \times line 7 \times line 52
57. $C_T = \text{line 53} + \text{line 54} + \text{line 55} + \text{line 56} - \text{line 22}$ _____ \$

Total Cost for Non-Solar System

58. $C_{oc} E_o = \text{line 32} \times \text{line 50}$ _____ \$
59. $Lc_{fc} E_f = \text{line 35} \times \text{line 7} \times \text{line 52}$ _____ \$
(maintenance cost neglected)
60. $C_{TC} = \text{line 58} + \text{line 59}$ _____ \$

Present Value of Life-Cycle Cost Savings
With Solar System

61. Savings = (line 60 - line 57) _____ \$

WORKSHEET LCA-3

In Worksheet LCA-3, column [1] is the year into the future for which the analysis may be made.

Column [2] is the annual mortgage payment (see Worksheet LCA-1, line 25). If the mortgage payment is a fixed annual amount, the payment for all future years would be the same as the first year.

Column [3] is the fraction of the mortgage payment which is paid as interest. The fraction decreases with increasing years and may be determined from Tables 12-14 through 12-18 for the particular interest rate and term of the mortgage.

Column [4] is the portion of the mortgage which is paid as interest and is the product of column [2] times column [3].

Column [5] is auxiliary fuel cost. Because fuel cost is expected to increase, the first year fuel cost should be increased in subsequent years. The first year fuel cost is the amount on line 26, Worksheet LCA-1. The second year fuel cost is determined by multiplying the first year cost by $(1 + r_f)$, (for r_f see line 12 of Worksheet LCA-1). For example, if the first year fuel cost is \$400 and the fuel inflation rate is 15 percent, the second year cost is $(\$400 \times 1.15 =)$ \$460. The fuel cost for each succeeding year is determined by multiplying the previous year by $(1 + \text{fuel inflation rate})$. Note that the inflation rate may be changed for any year in the period of analysis.

Column [6] is the annual property tax. The first year tax is calculated on line 27 on Worksheet LCA-1 and succeeding years can be escalated by the general inflation rate, g .

Column [7] is the annual insurance premium, which is determined on line 28 of Worksheet LCA-1 for the first year. Succeeding years may be

increased by a fixed or variable rate as desired. If no other guideline is available, insurance rates may be assumed to increase at the assumed general inflation rate.

Column [8] is the annual operating cost. The first year cost is estimated on line 29 of Worksheet LCA-1. Costs for succeeding years may be increased according to the fuel (electric) inflation rate.

Column [9] is the annual maintenance cost. The first year cost is on line 30 of LCA-1. Maintenance costs may be increased annually according to general inflation rates of items of replacement, such as motorized valves, pumps, or domestic hot water tanks. Such costs may be added in the future year when replacement is expected.

Column [10] is the income tax savings calculated by the product of the effective tax rate (on line 19 of Worksheet LCA-1) and the sum of annual interest paid, in column [4], plus property taxes, column [6].

Column [11] is the annual expense of solar system and is determined by: $\text{Column [2]} + \text{column [5]} + \text{column [6]} + \text{column [7]} + \text{column [8]} + \text{column [9]} - \text{column [10]}$. The first year cash flow is calculated by adding the down payment and subtracting the Federal tax credit.

LIFE-CYCLE COST ANALYSIS CASH FLOW

12-47

A. Mortgage interest rate _____ decimal Collector area _____ ft² System Cost \$ _____
 B. Auxiliary fuel inflation rate _____ decimal Solar fraction of total load _____ decimal Down Payment \$ _____
 C. General inflation rate _____ decimal (see Worksheet LCA-1, line 5) Federal Tax Credit \$ _____

[1] Year	[2] Annual Mortgage Payment	[3] Frac. of Mortgage as Interest	[4] Interest Paid	[5] Auxiliary Fuel Cost	[6] Property Tax	[7] Insurance	[8] Operating Cost	[9] Maintenance Cost	[10] Income Tax Savings	[11] Expense with Solar
1										*
2										
3										
4										
5										
6										
7										
8										
9										
10										
11										
12										
13										
14										
15										
16										
17										
18										
19										
20										

- [2] Annual mortgage payment from LCA-1, line 25
 [3] See Tables 12-14 through 12-18
 [4] Column [2] x column [3]
 [5] First year cost from LCA-1, line 26
 Second and future years:
 (previous year cost) x (1 + fuel inflation rate)
 See line 27, LCA-1
 Second and future years:
 (previous year cost) x (1 + general inflation rate)
 See line 28, LCA-1 (and use general inflation rate)
 [6] See line 27, LCA-1
 Second and future years:
 (previous year cost) x (1 + general inflation rate)
 See line 28, LCA-1 (and use general inflation rate)
 [7] See line 28, LCA-1 (and use general inflation rate)
 [8] First year cost see line 29, LCA-1
 Second and future years:
 (previous year cost) x (1 + fuel inflation rate)
 First year cost see line 30, LCA-1
 Second and future years:
 (previous year cost) x (1 + general inflation rate)
 {Column [4] + column [6]} x line 19, LCA-1
 [10] [2] + [5] + [6] + [7] + [8] + [9] - [10]
 [11] For first year, add down payment and subtract Federal tax credit (negative number indicates savings)
 * [11]

WORKSHEET LCA-4

Worksheet LCA-4 is used to compare life-cycle and cash-flow analyses for solar and non-solar systems.

Column [1] is the year of analysis.

Column [2] is the total fuel and operating cost for the non-solar system. The first year cost is the total of lines 31 and 32 on Worksheet LCA-1. The costs in succeeding years are determined by multiplying the cost of fuel for the previous year $(1 + r_f)$ and the cost for operation for the previous year by $(1 + r_e)$.

Column [3] is the cumulative annual cash flow for the non-solar system.

Column [4] is the present worth factor, determined from Table 12-7.

Column [5] is the present worth of the annual cost for a non-solar system.

Column [6] is the annual cost of the solar system, transferred from column [11] of Worksheet LCA-3.

Column [7] is the cumulative annual cash flow for the solar system.

Column [8] is the present value of the annual cost of the solar system determined by multiplying column [6] by column [4].

Column [9] is the present worth of savings with a solar system and is determined by column [5] - column [8].

Column [10] is the cumulative present worth of savings with a solar system and is the running sum of column [9].

Column [11] is the cumulative savings in cash flow and is determined by column [3] - column [7].

LIFE-CYCLE COST ANALYSIS
CASH FLOW AND PRESENT WORTH SUMMARIES

[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]	[9]	[10]	[11]
Year	NON-SOLAR SYSTEM			SOLAR SYSTEM						
	d = _____			Collector Area _____ ft ²						
	Fuel plus Operating Expenses	Cumulative Expenses	Present Worth Factor	Present Worth of Annual Cost	Expense with Solar System	Cumulative Expenses	Present Worth of Annual Cost	Present Worth of Savings	Cumulative Present Worth of Savings	Cumulative Savings (cash flow)
1										
2										
3										
4										
5										
6										
7										
8										
9										
10										
11										
12										
13										
14										
15										
16										
17										
18										
19										
20										

- [2] First year cost, add lines 31 and 32 of LCA-1
Second and future years:
(previous year cost) x (1 + fuel inflation rate)
- [3] Accumulate column [2]
- [4] See Table 12-7
- [5] Column [2] x column [4]
- [6] Column [11], Worksheet LCA-3
- [7] Accumulate column [6]
- [8] Column [6] x column [4]
- [9] Column [5] - column [8]
- [10] Running sum of column [9]
- [11] Column [3] - column [7]

EXAMPLE 12-4

Determine the life-cycle cost and energy cost savings of a liquid-heating solar system 500 square feet of collectors. Assume the following data apply:

1. F , annual solar fraction is 0.68.
2. Parasitic energy requirement is 7.5 percent of the solar energy collected.
3. c_e , current electrical energy cost is 5.0 ¢/kWh.
4. A 25-year loan at 10 percent is obtainable with 20 percent down.
5. Cost of solar system is \$17.25/ft² for collector-related costs plus \$7,285 for costs not related to collector area. In terms of collector area, the cost is $\$17.25 + (7285/500)$ or 31.82 \$/ft² of collector.
6. r_f , fuel inflation rate is 15 percent.
7. g , general inflation rate is 7 percent.
8. Homeowners insurance is available for a premium of 0.3 percent of insured value.
9. Property tax is levied on an assessed value which is 30 percent of market and the mil levy is 100. This is equivalent to a property tax levied as (0.30×0.1) , or 3 percent of market value.
10. The owner's Federal income tax rate is 32 percent and the state tax is 8 percent.
11. Maintenance cost is \$100 for first year.
12. Use a market discount rate of 10 percent.

DATA SHEET FOR ECONOMIC ANALYSIS

Project Sunbody Residence

Building Data

1. Annual space heating load 108.3 MMBtu/yr
2. Annual DHW heating load 21.0 MMBtu/yr
3. Total H and DHW load (add lines 1 & 2) 129.3 MMBtu/yr

Solar System Data

4. Collector area 500 ft²
5. Fraction of annual heating load supplied from solar 0.68 decimal

Energy Prices

6. c_e , current energy cost for electricity (use Figure 12-2) 5.0 ¢/kWh 14.65 \$/MMBtu
7. c_f , c_{fc} , current cost of fuel (oil) (use Figure 12-1 or 12-2) 12.00 \$/MMBtu

Terms of Loan

8. m , term of the loan for solar system 25 yrs
9. α , downpayment 20 % 0.20 decimal
10. i , interest rate on loan 10 % 0.10 decimal

Economic Data

11. C_a , installed cost of solar system per unit area 31.82 \$/ft²
12. r_f , estimated auxiliary fuel inflation rate 15 %
13. r_e , r_o , estimated electric energy inflation 15 %
14. g , r_m , estimated general inflation rate 7 %
15. p , property tax rate (based on market value) 0.03 decimal
16. h , insurance premium rate 0.003 decimal
17. Federal income tax rate for owner 0.32 decimal
18. State income tax rate for owner 0.08 decimal
19. t , effective income tax rate
{i.e., (line 17) + (line 18)
- [2 x (line 17) x (line 18)]} 0.35 decimal
20. d , market discount rate 0.10 decimal

Solar System Cost Items

- | | |
|---|-------------------|
| 21. Installed cost (line 4 x line 11) | <u>15,910</u> \$ |
| 22. Federal tax credit for solar
(40% of first \$10,000 of
system cost) | <u>4000</u> \$ |
| 23. Downpayment (line 21 x line 9) | <u>3182</u> \$ |
| 24. Amount of loan (line 21 - line 23) | <u>12,728</u> \$ |
| 25. Annual mortgage payment (multiply line
24 by annual mortgage rate from
Figure 12-4) | <u>1400</u> \$/yr |
| 26. C_f , first year cost of auxiliary heating
(line 3 x (1-line 5) x line 7) | <u>497</u> \$/yr |
| 27. First year property tax (line 21 x
line 15) | <u>477</u> \$/yr |
| 28. First year insurance premium
(line 21 x line 16) | <u>48</u> \$/yr |
| 29. C_o , first year cost of operating the
solar system (line 3 x (a value
between .05 and .10) x line 6) | <u>142</u> \$/yr |
| 30. C_m , first year maintenance cost
(estimate) | <u>100</u> \$/yr |

Non-Solar System Cost Items

- | | |
|--|-------------------|
| 31. C_{fc} , first year cost of fuel for non-
solar system (line 3 x line 7) | <u>1552</u> \$/yr |
| 32. C_{oc} , first year cost of operating
non-solar system (line 3 x
.01 x line 6) | <u>19</u> \$/yr |

LIFE-CYCLE COST ANALYSIS

Total Cost for Solar System

33. n, total years of analysis 20 yrs
34. A, collector area (line 4 of LCA-1) 500 ft²
35. L, annual heat load (line 3 of LCA-1) 129.3 MMBtu
36. F, fraction of annual heat provided
by the solar system (line 5 of
LCA-1) .68 decimal
37. P/X (d,g,n) (See Tables 12-1
through 12-6) 14.160
38. P/X (d,0,m) (See Tables 12-1
through 12-6) 9.077
39. P/X (i,0,m) (See Tables 12-1
through 12-6) 9.077
40. P/X (d,i,m) (See Tables 12-1
through 12-6) 22.727
41. P/X (0,i,m) (see Tables 12-1
through 12-6) 98.347
42. $(t) \left[\frac{P/X (d,i,m)}{P/X (0,i,m)} \right] = \left(\frac{\text{line 19} \times \text{line 40}}{\text{line 41}} \right)$ 0.081
43. $(1 - t) \left[\frac{P/X (d,0,m)}{P/X (i,0,m)} \right] = \left[\frac{(1 - \text{line 19}) \times (\text{line 38})}{\text{line 41}} \right]$ 0.650
44. Add line 42 and line 43 0.731
45. $1 - \alpha (1 - \text{line 9})$ 0.8
46. Multiply: line 44 x line 45 0.585
47. $(1-t)(p) + h$
 $(1 - \text{line 19})(\text{line 15}) + (\text{line 16})$ 0.0225
48. Multiply: line 47 x line 37 0.319
49. $E_1 = (\text{line 9}) + (\text{line 48}) + (\text{line 46})$ 1.104
50. $E_0 = P/X (d,r_0,n)$ (see Tables
12-1 through 12-6) 28.656
51. $E_m = P/X (d,r_m,n)$ (see Tables 12-1
through 12-6) 14.160

52. $E_f = P/X (d, r_f, n)$ (See Tables 12-1 through 12-6) 28.656
53. $(A)(C_a)(E_1) = (\text{line 34} \times \text{line 11} \times \text{line 49})$ 17,565 \$
54. $C_o E_o = (\text{line 29} \times \text{line 50})$ 4069 \$
55. $C_m E_m = (\text{line 30} \times \text{line 51})$ 1416 \$
56. $(1-F)(L)(c_f)(E_f) = (.32)(129.3)(12.00)(28.656)$ 14,228 \$
(1 - line 36) x line 35 x line 7 x line 52
57. $C_T = \text{line 53} + \text{line 54} + \text{line 55} + \text{line 56} - \text{line 22}$

<u>33,278 \$</u>

Total Cost for Non-Solar System

58. $C_{oc} E_o = \text{line 32} \times \text{line 50}$ 544 \$
59. $Lc_{fc} E_f = \text{line 35} \times \text{line 7} \times \text{line 52}$ 44,463 \$
(maintenance cost neglected)
60. $C_{TC} = \text{line 58} + \text{line 59}$ 45,007 \$

Present Value of Life-Cycle Cost Savings
With Solar System

61. Savings = (line 60 - line 57) 11,729 \$

EXAMPLE 12-5

Determine the life-cycle cost over 20 years of an air-heating solar space and water-heating system with 400 square feet of collectors. Calculate both the cumulative savings in cash flow and present worth of cumulative savings over the 20 years. Assume the following:

1. F , annual solar fraction is 0.50.
2. Parasitic energy requirement is 7.5 percent of the solar energy collected to store and distribute the heat.
3. c_e , current electrical energy cost is 5¢/kWh.
4. A 30-year loan at 10 percent is obtainable with 10 percent down-payment.
5. Cost of the solar system is \$17.25/ft² for collector-related costs plus \$7805 for costs not related to collector area. In terms of collector area, the cost is $[17.25 + (7805/400)] = \$36.76/\text{ft}^2$.
6. r_f , full inflation rate will vary, and let us assume a 40 percent increase over the next 2 years, dropping to 25 percent for the following 2 years, 15 percent for the following 4 years, and 8 percent for the balance of the years to 20 years.
7. g , general inflation rate will be affected by the fuel inflation rate and will tend to follow its pattern. Assume 12 percent for 2 years, 10 percent for the next 2 years, 8 percent for the following 4 years, and 6 percent for the balance of the years.
8. Homeowner's insurance is available for a premium of 0.3 percent of insured value for the first year (use system cost) and will increase according to the general inflation rate.
9. Property tax for the first year is 3 percent of market value and will increase steadily at 7 percent per year.
10. The owner's Federal income tax rate is 25 percent now but will increase to 45 percent over the 20 years of analysis. Assume a linear rate of increase, i.e. one percent per year. There is no state income tax.
11. Maintenance cost will be \$100 per year for the first 5 years, which will increase to \$150 for the following 5 years. In the 11th year there will be a motor replacement necessary which will cost \$300, but thereafter the maintenance cost remains at \$150/yr.

12. Assume a steady market discount rate of 10 percent over the 20-year period.

The annual cash flow for the solar system is calculated on Worksheet LCA-3. If the inflation scenario is realistic, the cost for heating the building in the year 2000 can be 12 times larger than present for the conventional heating system and 4 to 5 times larger with the solar plus auxiliary system.

According to the calculations, by the third year, the cost of heating with the solar plus auxiliary system is about the same as heating with auxiliary alone and with the solar system there is a 50 percent reduction in consumption of fuel oil. This observation is borne out in the calculations shown on Worksheet LCA-4, where cumulative present worth of savings over twenty years is calculated, and for comparison, cumulative savings of annual cash flow are also shown. The cumulative savings of cash flow is not a realistic value because, although "dollar" savings are added each year to determine the total savings, the value of the dollar changes each year because of inflation. The cumulative present worth column sums devalued dollars, and provided a realistic economic scenario has been applied, the owner of this solar house can expect to save a significant amount of money each year beyond the third year by installing the solar system. At the same time, consumption of heating fuel is reduced by about one-half.

DATA SHEET FOR ECONOMIC ANALYSIS

Project

Solar II

Building Data

1. Annual space heating load 108.3 MMBtu/yr
2. Annual DHW heating load 21.0 MMBtu/yr
3. Total H and DHW load (add lines 1 & 2) 129.3 MMBtu/yr

Solar System Data

4. Collector area 400 ft²
5. Fraction of annual heating load supplied from solar 0.50 decimal

Energy Prices

6. c_e , current energy cost for electricity (use Figure 12-2) 5 ¢/kWh 14.65 \$/MMBtu
7. c_f, c_{fc} , current cost of fuel (oil) (use Figure 12-1 or 12-2) 12.00 \$/MMBtu

Terms of Loan

8. m , term of the loan for solar system 30 yrs
9. α , downpayment 10 % 0.10 decimal
10. i , interest rate on loan 10 % 0.10 decimal

Economic Data

11. C_a , installed cost of solar system per unit area 36.76 \$/ft²
12. r_f , estimated auxiliary fuel inflation rate varies %
13. r_e, r_o , estimated electric energy inflation varies %
14. g, r_m , estimated general inflation rate varies %
15. p , property tax rate (based on market value) .03 decimal
16. h , insurance premium rate .003 decimal
17. Federal income tax rate for owner 0.25 decimal
18. State income tax rate for owner 0 decimal
19. t , effective income tax rate
{i.e., (line 17) + (line 18)
- [2 x (line 17) x (line 18)]} 0.25 decimal
20. d , market discount rate 0.10 decimal

Solar System Cost Items

21. Installed cost (line 4 x line 11) 14,704 \$
22. Federal tax credit for solar
(40% of first \$10,000 of
system cost) 4,000 \$
23. Downpayment (line 21 x line 9) 1,470 \$
24. Amount of loan (line 21 - line 23) 13,234 \$
25. Annual mortgage payment (multiply line
24 by annual mortgage rate from
Figure 12-4) 1403 \$/yr
26. C_f , first year cost of auxiliary heating
(line 3 x (1-line 5) x line 7) 776 \$/yr
27. First year property tax (line 21 x
line 15) 441 \$/yr
28. First year insurance premium
(line 21 x line 16) 44 \$/yr
29. C_o , first year cost of operating the
solar system (line 3 x (a value
between .05 and .10) x line 6) 24 \$/yr
30. C_m , first year maintenance cost
(estimate) 100 \$/yr

Non-Solar System Cost Items

31. C_{fc} , first year cost of fuel for non-
solar system (line 3 x line 7) 1552 \$/yr
32. C_{oc} , first year cost of operating
non-solar system (line 3 x
.01 x line 6) 19 \$/yr

LIFE-CYCLE COST ANALYSIS CASH FLOW

A. Mortgage interest rate .10 decimal Collector area 400 ft² System Cost \$ 4,704
 B. Auxiliary fuel inflation rate Var decimal Solar fraction of total load 0.5 decimal Down Payment \$ 1,470
 C. General inflation rate Var decimal (see Worksheet LCA-1, line 5) Federal Tax Credit \$ 4000

[1] Year	[2] Annual Mortgage Payment	[3] Frac. of Mortgage as Interest	[4] Interest Paid	[5] Auxiliary Fuel Cost	[6] Property Tax	[7] Insurance	[8] Operating Cost	[9] Maintenance Cost	[10] Income Tax Savings	[11] Expense with Solar
1	1403	.943	1323	776	441	44	24	100	441	-183 *
2	1403	.937	1315	1086	472	49	34	100	465	2679
3	1403	.931	1306	1521	505	55	47	100	489	3142
4	1403	.924	1296	1901	540	61	59	100	514	3550
5	1403	.916	1285	2377	578	67	74	100	549	4050
6	1403	.908	1274	2733	619	72	85	150	577	4485
7	1403	.898	1260	3143	662	78	97	150	596	4937
8	1403	.888	1246	3614	708	84	112	150	625	5446
9	1403	.877	1230	4157	758	91	129	150	656	6032
10	1403	.865	1214	4484	811	96	139	150	689	6399
11	1403	.851	1194	4848	868	102	150	300	722	6949
12	1403	.836	1173	5286	928	108	162	150	756	7281
13	1403	.820	1150	5655	993	115	175	150	793	7698
14	1403	.802	1125	6107	1063	122	189	150	831	8203
15	1403	.782	1097	6596	1137	129	204	150	871	8748
16	1403	.761	1068	7124	1217	137	220	150	914	9337
17	1403	.737	1034	7693	1302	145	238	150	958	9973
18	1403	.710	996	8309	1343	154	257	150	1003	10663
19	1403	.681	955	8974	1491	163	278	150	1052	11407
20	1403	.650	912	9691	1595	172	300	150	1103	12208

[2] Annual mortgage payment from LCA-1, line 25

[3] See Tables 12-14 through 12-18

[4] Column [2] x column [3]

[5] First year cost from LCA-1, line 26

Second and future years:

[6] (previous year cost) x (1 + fuel inflation rate)

See line 27, LCA-1

Second and future years:

[7] (previous year cost) x (1 + general inflation rate)

See line 28, LCA-1 (and use general inflation rate)

[8] First year cost see line 29, LCA-1

Second and future years:

[9] (previous year cost) x (1 + fuel inflation rate)

First year cost see line 30, LCA-1

Second and future years:

[10] (previous year cost) x (1 + general inflation rate)

{Column [4] + column [6]} x line 19, LCA-1

[11] [21] + [5] + [6] + [7] + [8] + [9] - [10]

For first year, add down payment and subtract Federal

tax credit (negative number indicates savings)

LIFE-CYCLE COST ANALYSIS
CASH FLOW AND PRESENT WORTH SUMMARIES

[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]	[9]	[10]	[11]
NON-SOLAR SYSTEM					SOLAR SYSTEM					
Year	d = _____					Collector Area _____ ft ²				
	Fuel plus Operating Expenses	Cumulative Expenses	Present Worth Factor	Present Worth Annual Cost	Expense with Solar System	Cumulative Expenses	Present Worth of Annual Cost	Present Worth of Savings	Cumulative Present Worth of Savings	Cumulative Savings (cash flow)
1	1571	1571	.909	1428	-183	-183	-166	1594	1594	1754
2	2199	3770	.826	1816	2679	2496	2213	-397	1197	1274
3	3079	6849	.751	2312	3142	5638	2360	-48	1149	121
4	3849	10698	.683	2629	3550	9188	2425	204	1353	1510
5	4811	15509	.621	2988	4050	13238	2515	473	1826	2271
6	5533	21042	.564	3121	4485	17223	2530	591	2417	3319
7	6363	27405	.513	3264	4937	22660	2533	731	3148	4745
8	7317	34722	.467	3417	5446	28106	2543	814	4022	6616
9	8415	43137	.424	3568	6032	34138	2558	1010	5032	8499
10	9088	52225	.386	3508	6399	40537	2470	1038	6070	11688
11	9815	62040	.350	3435	6449	47486	2432	1003	7073	14554
12	10600	72640	.319	3381	7281	54767	2323	1058	8131	17873
13	11448	84088	.290	3320	7698	62465	2232	1088	9219	21623
14	12464	97052	.263	3410	8203	70668	2151	1253	10472	26384
15	13353	110405	.239	3191	8748	79416	2091	1100	11572	30989
16	14421	124826	.218	3144	9337	88753	2035	1109	12681	36073
17	15515	140401	.198	3084	9973	98726	1975	1109	13790	41675
18	16821	157222	.180	3028	10663	109389	1919	1109	14899	47833
19	18167	175389	.164	2979	11407	120796	1871	1108	16007	54593
20	19620	195009	.149	2923	12208	133004	1819	1104	17111	62005

[2] First year cost, add lines 31 and 32 of LCA-1

Second and future years:

(previous year cost) x (1 + fuel inflation rate)

[3] Accumulate column [2]

[4] See Table 12-7

[5] Column [2] x column [4]

[6] Column [11], Worksheet LCA-3

[7] Accumulate column [6]

[8] Column [6] x column [4]

[9] Column [5] - column [8]

[10] Running sum of column [9]

[11] Column [3] - column [7]

DATA SHEET FOR ECONOMIC ANALYSIS

Project _____

Building Data

1. Annual space heating load _____ MMBtu/yr
2. Annual DHW heating load _____ MMBtu/yr
3. Total H and DHW load (add lines 1 & 2) _____ MMBtu/yr

Solar System Data

4. Collector area _____ ft^2
5. Fraction of annual heating load
supplied from solar _____ decimal

Energy Prices

6. c_e , current energy cost for electricity
(use Figure 12-2) _____ $\text{\$/MMBtu}$
7. c_f, c_{fc} , current cost of fuel
(use Figure 12-1 or 12-2) _____ $\text{\$/MMBtu}$

Terms of Loan

8. m , term of the loan for solar system _____ yrs
9. α , downpayment _____ % _____ decimal
10. i , interest rate on loan _____ % _____ decimal

Economic Data

11. C_a , installed cost of solar system per
unit area _____ $\text{\$/ft}^2$
12. r_f , estimated auxiliary fuel inflation
rate _____ %
13. r_e, r_o , estimated electric energy
inflation _____ %
14. g, r_m , estimated general inflation
rate _____ %
15. p , property tax rate (based on
market value) _____ decimal
16. h , insurance premium rate _____ decimal
17. Federal income tax rate for owner _____ decimal
18. State income tax rate for owner _____ decimal
19. t , effective income tax rate
{i.e., (line 17) + (line 18)
- [2 x (line 17) x (line 18)]} _____ decimal
20. d , market discount rate _____ decimal



Solar System Cost Items

21. Installed cost (line 4 x line 11) _____ \$
22. Federal tax credit for solar
(40% of first \$10,000 of
system cost) _____ \$
23. Downpayment (line 21 x line 9) _____ \$
24. Amount of loan (line 21 - line 23) _____ \$
25. Annual mortgage payment (multiply line
24 by annual mortgage rate from
Figure 12-4) _____ \$/yr
26. C_f , first year cost of auxiliary heating
(line 3 x (1-line 5) x line 7) _____ \$/yr
27. First year property tax (line 21 x
line 15) _____ \$/yr
28. First year insurance premium
(line 21 x line 16) _____ \$/yr
29. C_o , first year cost of operating the
solar system (line 3 x (a value
between .05 and .10) x line 6) _____ \$/yr
30. C_m , first year maintenance cost
(estimate) _____ \$/yr

Non-Solar System Cost Items

31. C_{fc} , first year cost of fuel for non-
solar system (line 3 x line 7) _____ \$/yr
32. C_{oc} , first year cost of operating
non-solar system (line 3 x
.01 x line 6) _____ \$/yr



LIFE-CYCLE COST ANALYSIS

Total Cost for Solar System

33. n, total years of analysis _____ yrs
34. A, collector area (line 4 of LCA-1) _____ ft²
35. L, annual heat load (line 3 of LCA-1) _____ MMBtu
36. F, fraction of annual heat provided
by the solar system (line 5 of
LCA-1) _____ decimal
37. P/X (d,g,n) (See Tables 12-1
through 12-6) _____
38. P/X (d,0,m) (See Tables 12-1
through 12-6) _____
39. P/X (i,0,m) (See Tables 12-1
through 12-6) _____
40. P/X (d,i,m) (See Tables 12-1
through 12-6) _____
41. P/X (0,i,m) (see Tables 12-1
through 12-6) _____
42. $(t) \left[\frac{P/X (d,i,m)}{P/X (0,i,m)} \right] = \left(\frac{\text{line 19} \times \text{line 40}}{\text{line 41}} \right)$ _____
43. $(1 - t) \left[\frac{P/X (d,0,m)}{P/X (i,0,m)} \right] = \left[\frac{(1 - \text{line 19}) \times (\text{line 38})}{\text{line 41}} \right]$ _____
44. Add line 42 and line 43 _____
45. $1 - \alpha (1 - \text{line 9})$ _____
46. Multiply: line 44 x line 45 _____
47. $(1-t)(p) + h$
 $(1 - \text{line 19})(\text{line 15}) + (\text{line 16})$ _____
48. Multiply: line 47 x line 37 _____
49. $E_1 = (\text{line 9}) + (\text{line 48}) + (\text{line 46})$ _____
50. $E_0 = P/X (d,r_0,n)$ (see Tables
12-1 through 12-6) _____
51. $E_m = P/X (d,r_m,n)$ (see Tables 12-1
through 12-6) _____



52. $E_f = P/X (d, r_f, n)$ (See Tables 12-1 through 12-6)

53. $(A)(C_a)(E_1) = (\text{line 34} \times \text{line 11} \times \text{line 49})$ _____ \$

54. $C_o E_o = (\text{line 29} \times \text{line 50})$ _____ \$

55. $C_m E_m = (\text{line 30} \times \text{line 51})$ _____ \$

56. $(1-F)(L)(c_f)(E_f) = (\text{line 36}) (\text{line 35}) (\text{line 7}) (\text{line 52})$ _____ \$

57. $C_T = \text{line 53} + \text{line 54} + \text{line 55} + \text{line 56} - \text{line 22}$

_____ \$

Total Cost for Non-Solar System

58. $C_{oc} E_o = \text{line 32} \times \text{line 50}$ _____ \$

59. $Lc_{fc} E_f = \text{line 35} \times \text{line 7} \times \text{line 52}$ _____ \$
(maintenance cost neglected)

60. $C_{TC} = \text{line 58} + \text{line 59}$ _____ \$

Present Value of Life-Cycle Cost Savings
With Solar System

61. Savings = (line 60 - line 57)

_____ \$



LIFE CYCLE COST ANALYSIS
CASH FLOW

A. Mortgage interest rate decimal

B. Auxiliary fuel inflation rate decimal

C. General inflation rate decimal

Collector area

Solar fraction of total load

(see Worksheet LCA-1, line 5)

ft²

decimal

System Cost \$

Down Payment \$

Federal Tax Credit \$

[1] Year	[2] Annual Mortgage Payment	[3] Frac. of Mortgage as Interest	[4] Interest Paid	[5] Auxiliary Fuel Cost	[6] Property Tax	[7] Insurance	[8] Operating Cost	[9] Maintenance Cost	[10] Income Tax Savings	[11] Expense with Solar
1										*
2										
3										
4										
5										
6										
7										
8										
9										
10										
11										
12										
13										
14										
15										
16										
17										
18										
19										
20										

[2] Annual mortgage payment from LCA-1, line 25

[3] See Tables 12-14 through 12-18

[4] Column [2] x column [3]

[5] First year cost from LCA-1, line 26

Second and future years:
(previous year cost) x (1 + fuel inflation rate)

[6] See line 27, LCA-1

Second and future years:
(previous year cost) x (1 + general inflation rate)

[7] See line 28, LCA-1 (and use general inflation rate)

[8] First year cost see line 29, LCA-1

Second and future years:
(previous year cost) x (1 + fuel inflation rate)

[9] First year cost see line 30, LCA-1

Second and future years:
(previous year cost) x (1 + general inflation rate)

[10] {Column [4] + column [6]} x line 19, LCA-1

[11] [2] + [5] + [6] + [7] + [8] + [9] - [10]

*[11] For first year, add down payment and subtract Federal tax credit (negative number indicates savings)

LIFE-CYCLE COST ANALYSIS
CASH FLOW AND PRESENT WORTH SUMMARIES

[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]	[9]	[10]	[11]
NON-SOLAR SYSTEM			SOLAR SYSTEM							
d = _____			Collector Area _____ ft ²							
Year	Fuel plus Operating Expenses	Cumulative Expenses	Present Worth Factor	Present Worth of Annual Cost	Expense with Solar System	Cumulative Expenses	Present Worth of Annual Cost	Present Worth of Savings	Cumulative Present Worth of Savings	Cumulative Savings (cash flow)
1										
2										
3										
4										
5										
6										
7										
8										
9										
10										
11										
12										
13										
14										
15										
16										
17										
18										
19										
20										

[2] First year cost, add lines 31 and 32 of LCA-1
Second and future years:
(previous year cost) x (1 + fuel inflation rate)
[3] Accumulate column [2]
[4] See Table 12-7
[5] Column [2] x column [4]
[6] Column [11], Worksheet LCA-3
[7] Accumulate column [6]
[8] Column [6] x column [4]
[9] Column [5] - column [8]
[10] Running sum of column [9]
[11] Column [3] - column [7]



APPENDIX

State Legislation

ALASKA

TAX INCENTIVES

Alaska allows a 10% residential fuel conservation credit of up to \$200 per individual or married couple for money spent on the following: 1) insulation; 2) insulating windows; 3) labor related to items 1 and 2; 4) alternate energy systems which are not dependent on fossil fuel, including solar, wind, tidal, and geothermal. Expires 12/31/82 (Chapter 94, Laws of 1977).

Contact State Department of Revenue
 Income Tax Division
 Pouch SA
 continue State Office Building
 Juneau, AK 99811
 (907) 465-2326

GRANTS AND LOANS

This act creates the Alaska Renewable Research Corporation. The Corporation is funded by a portion of the state's proceeds from mineral leases, rentals, bonuses, and royalties. Most of the Corporation's budget is used as venture capital for new businesses involved in renewable resources such as forest products, fisheries, agriculture and renewable energy resources. Up to 10% of every annual appropriation may be used as grants to the same types of enterprises (Chapter 179, Laws of 1978).

Contact Alaska Renewable Research Corporation
 Box 1647
 Juneau, AK 99802
 (907) 465-4616

 Suite One
 313 "E" Street
 Anchorage, AK 99501
 (907) 272-2500

This law establishes an Alternative Power Resource Revolving Loan Fund in the Department of Commerce and Economic Development. The fund shall be used to develop energy production from sources other than fossil or nuclear fuel. This includes wind, water and solar devices. Loans may not exceed \$10,000 (Chapter 29, Laws of 1978).

Contact Division of Business Loans
 Department of Commerce & Economic Development
 Pouch D
 Juneau, AK 99811
 (907) 465-2510
 (907) 274-6693 (Anchorage)
 (907) 452-8182 (Fairbanks)



3/30/80

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ARIZONA

TAX INCENTIVES

Individuals may claim an income tax credit equal to 35% of the cost of a solar energy device through 1983. Thereafter the percentage decreases 5% per year until the credit expires on 12/31/89. The maximum credit is \$1,000. Home builders may claim the credit on new speculative solar homes in lieu of the purchaser. The credit is available at the same rates for commercial and industrial solar installations. All credits include carry forward provisions (Chapter 93, Laws of 1975; Chapter 129, Laws of 1976; Chapter 112, Laws of 1978; Chapter 146, Laws of 1979).

Solar energy devices are exempt from property taxes through 12/31/89 (Chapter 165, Laws of 1974; Chapter 146, Laws of 1979). Solar energy devices are exempt from Transaction Privilege and Use Taxes through 12/31/89 (Chapter 42, Laws of 1977; Chapter 146, Laws of 1979). A 25% credit is allowed for residential insulation and ventilation devices, such as insulating doors and windows. Maximum credit is \$100. Credit expires 12/31/84 (Chapter 112, Laws of 1978).

Contact State Department of Revenue
 Box 29002
 Phoenix, AZ 85038
 (602) 255-3381

LAND USE

Provides authority for local governments to regulate solar access (Chapter 94, Laws of 1979).

Contact Local planning commission or zoning board

STANDARDS AND REGULATION OF CONSTRUCTION

Local building codes may include provisions requiring that new, single-family residences be designed to facilitate future installation of solar heating equipment (Chapter 94, Laws of 1979).

Contact Local building official

ARKANSAS

TAX INCENTIVES

An energy conservation income tax deduction is available to any individual, fiduciary or corporation. Eligible items include insulation and energy conservation adjustments made to buildings constructed before 1/1/79. Also eligible are these additions made to new or existing buildings: devices using solar energy, bioconversion energy, geothermal energy, hydroelectric energy (for mechanical or electrical power) for space heating or cooling or water heating; devices using wind energy; and devices creating energy from woodburning in stoves, furnaces, or fireplaces with controllable drafts and dampers. Deductions exceeding income may be carried forward until exhausted. Eligible expenditures must be made before 12/31/84. The Commissioner of Revenue may promulgate regulations for the administration of the deduction (Act 535, 1977; Act 742, 1979; Act 59, 1980).

Contact State Department of Revenue
 Income Tax Section
 7th and Wolfe Streets
 Little Rock, AR 72201
 (501) 371-2193

CALIFORNIA

TAX INCENTIVES

California provides personal income tax credit of 55% of the cost of a solar energy system, up to a maximum of \$3000. If a system is installed in other than a single-family dwelling and the cost exceeds \$6000, the credit equals 25% of the cost, or \$3000, whichever is greater. In both single-family dwellings and other buildings, the cost of energy-conserving devices installed in conjunction with the solar energy system may also be included in the total cost used to

calculate the tax credit. If a federal credit is claimed, the state credit is reduced by the amount of the federal credit. The same provisions apply to corporate taxpayers. Taxpayers who partially own and partially lease a solar system from a public utility are also eligible for the credit. The system must meet the criteria of the California Energy Commission. Eligible expenses for credit include attorney's fees, compensation, and recording fees associated with obtaining a solar easement. Credit expires 12/31/80 (Chapter 168, Laws of 1976; Chapter 1082, Laws of 1977; Chapter 1154, Laws of 1978; Chapter 816, Laws of 1979).

Contact Franchise Tax Board
Attn: Correspondence
Sacramento, CA 95807
(916) 355-0370

GRANTS AND LOANS

This law creates the Solar Energy Demonstration Loan Program and provides \$2000 interest-free loans for solar space heating and domestic hot water systems in areas where a state of emergency has been declared (Chapter 1 and Chapter 7, Laws of 1978).

Contact Department of Housing and Community Development
Division of Research and Policy Development
921 10th Street, Fifth Floor
Sacramento, CA 95814
(916) 445-4728

The maximum loan available to veterans for home mortgages is increased to \$60,000 if the home is equipped with a solar energy system (Chapter 1243, Laws of 1978; Chapter 48, Laws of 1979).

Contact Department of Veterans' Affairs
1227 "O" Street
P.O. Box 1559
Sacramento, CA 95807
(916) 445-2347
or any local district office of the
Department of Veterans' Affairs

LAND USE

Anyone who owns, occupies, or controls real estate is prohibited from allowing a tree or shrub to cast a shadow on a solar collector between 9:30 a.m. and 2:30 p.m. Trees casting a shadow before the installation of a collector are excluded (Chapter 1366, Laws of 1978).

This law declares that any restriction on real property purporting to prohibit the installation and use of a solar energy system is void and unenforceable. It recognizes solar easements and prescribes their contents. City and county governments may not prohibit or restrict solar energy systems except to ensure the public health. The law requires that new subdivision maps be designed to accommodate passive solar energy systems to the maximum extent possible. It permits city and county governments to require the dedication of solar easements before approving the map (Chapter 1154, Laws of 1978).

Contact Local planning or zoning body

STANDARDS AND REGULATION OF CONSTRUCTION

Any city or county may require that new buildings subject to the State Housing Law be constructed in a manner that permits the installation of solar heating (Chapter 670, Laws of 1976).

Contact Local building inspector

The California Energy Commission is required to adopt regulations and standards governing solar energy equipment (Chapter 1081, Laws of 1977).

Contact California Energy Commission
1111 Howe Avenue
Sacramento, CA 95825
(916) 322-3690

COLORADO

TAX INCENTIVES

Alternative energy devices will be assessed for real estate tax at 5% of their value. Eligible devices must use solar or geothermal energy, renewable biomass, or wind resources. Passive solar designs are included, but devices for the direct combustion of wood are ineligible. The exemption expires 12/31/89 (Chapter 344, Laws of 1975; Chapter 363, Laws of 1979).

Contact Local tax assessor

Creates an income tax deduction for alternative energy devices. Eligible deductions include the cost of solar, wind, geothermal, or renewable biomass systems. Passive solar designs are included to the extent that construction costs exceed those of conventional designs. Devices for the direct combustion of wood do not qualify. Corporate taxpayers may use the deduction in lieu of depreciation (Chapter 512, Laws of 1977; Chapter 374, Laws of 1979).

Contact Colorado Department of Revenue
Capitol Annex Building
1375 Sherman Street
Denver, CO 80203
(303) 839-2801

LAND USE

Solar easements are recognized and their contents are prescribed. They are subjected to the same conveyancing and recording requirements as other easements. Any unreasonable restriction on real estate, based on aesthetic considerations and effectively prohibiting or restricting the installation and use of a solar energy device, is declared void and unenforceable (Chapter 326, Laws of 1975; Chapter 358, Laws of 1979).

This law authorizes local governments to regulate uses of land in planning and zoning regulations to assure access to direct sunlight for solar energy devices. Special exceptions to zoning regulations may be granted to protect solar access. Subdivision regulations may be altered to protect solar access. Effective 1/1/80 (Chapter 306, Laws of 1979).

Contact Local zoning board or planning commission

CONNECTICUT

TAX INCENTIVES

Municipalities are authorized to exempt windmills, waterwheels and solar heating, cooling or electrical systems from real estate tax. Installation must occur before 10/1/91. The exemption will be effective for 15 years after installation. Passive solar energy systems, constructed or installed after 10/1/80 may also be exempted. Systems must meet standards of the Office of Policy and Management (Public Act 76-409; Public Act 77-490; Public Act 79-479).

Solar collectors are exempt from sales tax through 10/1/82 (Public Act 77-457)

Contact Commissioner of Revenue Services
92 Farmington Avenue
Hartford, CT 06115
(203) 566-7120

GRANTS AND LOANS

This law requires the Commissioner of Economic Development to establish an energy conservation loan fund. The Commissioner shall make low-cost loans from this fund for insulation, energy conservation materials and alternate energy devices to be installed in residential buildings containing not more than four dwelling units. Alternate energy devices are those which use solar radiation, wind, water, or geothermal resources for space heating or cooling, water heating or generation of electricity. Loans can range from \$400 to \$3000. The State Bond Commission may authorize the issuance of bonds to fund the loan program (Public Act 79-509).

The law authorizes the Department of Economic Development to make loans for industrial applications of energy conservation techniques, solar, wind, hydro, biomass or other forms of renewable energy (Public Act 79-520).

Contact Department of Economic Development
210 Washington Street
Hartford, CT 06115
(203) 566-4555

This law authorizes the Connecticut Housing Financing Authority to make and insure loans for energy conservation improvements and installation of renewable energy systems for space heating and cooling, water heating, and electricity in residential buildings. Renewable energy sources eligible include wind, solar, water and biomass (Public Act 79-578).

Contact Connecticut Housing Financing Authority
190 Trumbull Street
Hartford, CT 06103
(203) 525-9311

LAND USE

The zoning commission of each city, town, or borough is authorized to regulate development to encourage energy efficiency and the use of renewable forms of energy, including solar (Public Act 78-314).

Contact Local planning or zoning body

STANDARDS AND REGULATION OF CONSTRUCTION

The Office of Policy and Management is required to establish standards for solar energy systems (Public Act 76-409; Public Act 79-479).

Contact Office of Policy and Management
20 Grand Street
Hartford, CT 06115
(203) 566-5765

DELAWARE

TAX INCENTIVES

This law provides an income tax credit of \$200 for solar energy devices designed to produce domestic hot water. Systems must meet HUD Intermediate Minimum Property Standards Supplement for Solar Heating and Domestic Hot Water Systems. Systems must be warranted according to criteria set out in the law (Chapter 512, Laws of 1978).

Contact Division of Revenue
State Office Building
820 French Street
Wilmington, DE 19801
(302) 571-3360

FLORIDA

TAX INCENTIVES

Effective 7/1/79, solar energy systems are exempt from sales and use tax until 6/30/84 (Chapter 339, Laws of 1979).

Contact Florida Department of Revenue
Sales Tax Division
Carlton Building
Tallahassee, FL 32304
(904) 488-6800

LAND USE

Solar easements are recognized and subject to the same requirements as other easements; the contents are prescribed (Chapter 309, Laws of 1978).

STANDARDS AND REGULATION OF CONSTRUCTION

The Florida Solar Energy Center is required to establish standards for solar energy systems (Chapter 246, Laws of 1976).

Contact Florida Solar Energy Center
300 State Road 401
Cape Canaveral, FL 32920
(305) 783-0300

The law stipulates that no single-family dwelling shall be constructed unless it is designed to facilitate future installation of a solar hot water system (Chapter 361, Laws of 1974).

Contact Local building inspector

All solar energy systems manufactured or sold in Florida must meet the standards established by the Florida Solar Energy Center (Chapter 309, Laws of 1978).

Contact Bureau of Codes and Standards
2571 Executive Center Circle
Tallahassee, FL 32301
(904) 488-3581

GEORGIA

TAX INCENTIVES

Real estate owners may claim a refund of sales tax paid for the purchase of solar equipment. Expires 7/1/86 (Act 1030, 1976; Act 1309, 1978).

Contact State Department of Revenue, Sales Tax Division
309 Trinity-Washington Building
Atlanta, GA 30334
(404) 656-4065

Any county or municipality may exempt solar heating and cooling equipment and machinery used to manufacture solar equipment from property taxes. Expires 7/1/86 (Georgia Constitution, Article VII, Section 1, Paragraph IV).

Contact Local city council or county board of supervisors

LAND USE

Solar easements are recognized and subject to the same requirements as other easements; the contents are prescribed (Act 1446, 1978).

HAWAII

TAX INCENTIVES

A 10% income tax credit is provided to individuals and corporations who purchase solar energy devices that are placed in service by 12/31/81. The law also provides property tax exemptions for solar energy systems through 12/31/81. This exemption also applies to any non-nuclear and non-fossil fuel system and to any improvement that increases the efficiency of systems which use fossil fuel (Act 189, 1976).

Contact State Tax Department
P.O. Box 259
Honolulu, HI 96809
(808) 548-3270

IDAHO

TAX INCENTIVES

This law allows an income tax deduction for a solar heating/cooling or solar electrical system installed in the taxpayer's residence. The deduction equals 40% of the cost in the first year and 20% of the cost in each of the next 3 years; the maximum deduction in any year is \$5000. This deduction also applies to systems fueled by wind, geothermal energy, wood, or wood products. Built-in fireplaces qualify if they have control doors, regulated draft, and heat exchangers that deliver heated air to substantial portions of the residence (Chapter 212, Laws of 1976).

Contact State Tax Commission
5257 Fairview
Boise, ID 83722
(208) 384-3290

LAND USE

Solar easements are recognized and are made subject to the same requirements as other easements; the contents are prescribed (Chapter 294, Laws of 1978.)

ILLINOIS

TAX INCENTIVES

A property owner who installs a solar or wind energy system may claim an alternate valuation for property taxes. The property is assessed twice: with the solar or wind energy system and also as though it were equipped with a conventional system. The lesser of the two assessments is used to compute the tax due. Owners must file a claim with the local Board of Assessors. (Public Act 79-943, 1975; Public Act 80-430, 1977).

Contact Local assessor or board of assessors

GRANTS AND LOANS

This law establishes a \$5 million research, development, and demonstration program for non-coal, non-nuclear energy (Public Act 80-432, 1977).

Contact Illinois Institute of Natural Resources
325 West Adams
Room 300
Springfield, IL 62706
(217) 785-2800

STANDARDS AND REGULATION OF CONSTRUCTION

The Illinois Institute of Natural Resources is required to establish guidelines and regulations for solar energy systems (Public Act 80-430, 1977).

Contact Illinois Institute of Natural Resources
325 West Adams
Room 300
Springfield, IL 62706
(217) 785-2800

INDIANA

TAX INCENTIVES

The law permits the property owner who installs a solar heating and cooling system to have property assessment reduced by the difference between the assessment of the property with the system and the assessment of the property without the system. The owner must apply to the county auditor (Public Law 15, 1974; Public Law 68, 1977).

Contact Local assessor or board of assessors

An income tax credit is created for individuals, corporations, and partnerships that install solar or wind energy systems for heating space or water or for generating electricity. For single-family dwellings the credit equals 25% of eligible expenditures to a maximum of \$3,000. For other buildings the credit equals 25% of expenditures to a maximum of \$10,000. Credit exceeding tax liability may be carried forward until exhausted. Expenditures must be made by 12/31/82. The Department of Revenue shall promulgate rules to implement the credit, including performance and quality standards for qualifying systems. (Public Law 20, 1980).

Contact Indiana Department of Revenue
202 State Office Building
Indianapolis, IN 46204
(317) 232-2101

IOWA

TAX INCENTIVES

Installation of a solar energy system will not increase the assessed, actual, or taxable values of property for 1979-1985 (Section 441.21, Code of 1979).

Contact Local assessor or board of assessors

GRANTS AND LOANS

This law establishes a loan and grant fund for property improvements and mortgages for low-income families. Solar energy systems qualify as improvements (Chapter 1086, Laws of 1978).

Contact Iowa Housing Finance Authority
218 Liberty Building
Des Moines, IA 50319
(515) 281-4058

KANSAS

TAX INCENTIVES

The individual taxpayer is allowed an income tax credit of 25% of the cost of a residential solar energy system to a maximum of \$1000. A solar energy installation on business or investment property receives a credit equal to 25% of the system cost, \$3000, or that year's tax bill, whichever is the least amount. The cost of an installation on business or investment property can be amortized over 60 months. Wind energy systems are also covered by this law. Credit expires 7/1/83. (Chapter 434, Laws of 1976; Chapter 346, Laws of 1977).

If a solar system supplies 70% of the energy for heating and cooling, the property owner may be reimbursed for 35% of his property tax for up to 5 consecutive years. Applies through 1985. Claims must be filed with the Department of Revenue (Chapter 345, Laws of 1977; Chapter 419, Laws of 1978).

Provides an income tax deduction of 50% (maximum \$500) of the cost of insulating residential buildings owned by the taxpayer. The deduction can be applied separately to each residential building owned and insulated by the taxpayer. To qualify, a building must have been constructed before 7/1/77. The insulating materials must meet minimum standards for energy conservation in new buildings prescribed by the Federal Housing Administration (Chapter 410, Laws of 1978).

Contact State Department of Revenue
P.O. Box 692
Topeka, KS 66601
(913) 296-3909

LAND USE

Solar easements are recognized and are subject to the same requirements as other easements; the contents are prescribed (Chapter 277, Laws of 1977).

LOUISIANA

TAX INCENTIVES

Solar energy equipment installed in owner-occupied residential buildings or in swimming pools are exempt from property tax (Act 591, 1978).

Contact Local parish tax assessor

STANDARDS AND REGULATION OF CONSTRUCTION

The law requires the Department of Natural Resources and Development to adopt regulations and standards governing solar energy devices (Act 542, 1978).

Contact Department of Natural Resources and Development
P.O. Box 44396
Baton Rouge, LA 70804
(504) 342-4500

MAINE

TAX INCENTIVES

Solar space or water heating systems are exempt from property tax for 5 years after installation. Eligible taxpayers must apply to the local Board of Assessors. Purchasers of solar energy systems may also receive a sales tax rebate from the Office of Energy Resources (Chapter 542, Laws of 1977).

Contact (for property tax) Local assessor or board of assessors.

Contact (for sales tax) Office of Energy Resources
55 Capitol Street
Augusta, ME 04330
(207) 289-3811

This law creates an income tax credit for solar, wind, and wood energy systems which provide space or water heating or electrical or mechanical power. Fireplaces and woodstoves not operating as central heating systems are ineligible. Both active and passive solar systems qualify. The credit equals the lesser of \$100 or 20% of eligible expenditures. Retroactive to 1/1/79 (Chapter 557, Laws of 1979).

Contact Bureau of Taxation
Department of Finance and Administration
State Office Building
Augusta, ME 04333
(207) 289-2076

LAND USE

Local governments are permitted to enact zoning ordinances to protect access to direct sunlight for solar energy use (Chapter 418, Laws of 1979).

Contact Local zoning or planning group

Planning boards are permitted to protect solar access in new developments through subdivision regulations. These may include restrictive covenants, height restrictions, and setback requirements (Chapter 435, Laws of 1979).

Contact Local planning board

MARYLAND

TAX INCENTIVES

A solar energy unit will be assessed at no more than a conventional system needed to serve the building (Chapter 509, Laws of 1975; Chapter 509, Laws of 1978).

Contact Local assessor or board of assessors

Baltimore City and any other city or county may offer property tax credits for the use of solar systems in any type of building. Credit may be applied over a 3-year period (Chapter 740, Laws of 1976).

Contact Local city or county department of revenue

LAND USE

Solar easements are recognized as a lawful restriction on land (Chapter 934, Laws of 1977).

GRANTS AND LOANS

This law authorizes the city of Baltimore to issue \$2 million in municipal bonds. The proceeds from the sale of bonds shall be used for energy conservation loans and loan guarantees to improve residential buildings in the city. The bond issue requires an ordinance of the Baltimore City Council and the approval of the electorate of the city (Chapter 12, Laws of 1979).

MASSACHUSETTS

TAX INCENTIVES

Solar energy systems are exempt from property tax for 20 years from the date of installation (Chapter 734, Laws of 1975; Chapter 388, Laws of 1978).

Contact Local assessor or board of assessors

A personal income tax credit of 35% of the cost of renewable energy equipment is created. The equipment must be installed in the taxpayer's principal residence in the state. Maximum credit is \$1000. Eligible equipment must use solar energy for space heating or cooling or water heating or must use wind energy for any nonbusiness residential purpose. If a federal income tax credit or grant is received by the taxpayer, the state credit will be reduced. The credit expires 12/31/83.

Sales of equipment for residential solar energy systems, wind power systems, or heat pumps are exempt from sales tax. Wood-fueled central heating systems installed in a person's principal residence in the state and costing more than \$900 are exempt from sales tax through 12/31/83. Eligible furnaces must be approved by the State Fire Marshall or Building Code Commissioner (Chapter 796, Laws of 1979).

Corporations may deduct the cost of a solar or wind energy system from income. The system will also be exempt from tangible property tax (Chapter 487, Laws of 1977).

Contact State Department of Corporations & Taxation
100 Cambridge Street
Boston MA 02204
(617) 727-4201 (income tax)
(617) 727-4601 (sales tax)

GRANTS AND LOANS

Banks and credit unions are authorized to make loans with extended maturation periods and increased maximum amounts for home improvements, including solar energy systems. Banks may lend up to \$15,000, and credit unions may lend up to \$12,000 (Chapter 28, Laws of 1977; Chapter 260, Laws of 1977; Chapter 73, Laws of 1978).

Contact Local bank or credit union

MICHIGAN

TAX INCENTIVES

This law exempts solar, wind, or water energy conversion devices from real and personal property tax. An application must be filed with local tax assessor, who will submit it to the state tax commission for certification. Authority to exempt expires 7/1/85, but exemptions made by that time stay in force (Public Act 135, 1976).

Contact Local Government Services
 Treasury Building
 Lansing, MI 48922
 (517) 373-3232

Proceeds from sales of solar, wind, or water energy conversion devices used for heating, cooling, or electrical generation in new or existing residential or commercial buildings are excluded from business activities tax. Expires 1/1/85 (Public Act 132, 1976).

Tangible property used for solar, wind, or water energy devices is excluded from excise tax if it is used to heat, cool, or electrify a new or existing commercial or residential building. Expires 1/1/85 (Public Act 133, 1976).

Income tax credit may be claimed for a residential solar, wind, or water energy device that is used for heating, cooling, or electricity. This includes devices designed to use the difference between water temperatures in a body of water. Energy conservation measures installed in connection with such devices are also eligible; these include insulation, water-flow reduction devices, and some wood furnaces. Swimming pool heaters are eligible only if 25% or more of their heating capacity is used for residential purposes. The credit is refundable. The law instructs the Department of Commerce to establish system eligibility standards within 180 days of the law's passage. To be eligible, expenditures must be made by 12/31/83. The rate of credit changes annually. For 1980, the rate for single-family dwellings is 25% of the first \$2000 spent, plus 15% of the next \$8000 spent. In 1980, the rate for other buildings is 25% of the first \$2000, plus 15% of next \$13,000 (Public Act 605, 1978; Public Act 41, 1979).

Contact State Department of Treasury
 State Tax Commission
 State Capitol Building
 Lansing, MI 48922
 (517) 373-2910

STANDARDS AND REGULATION OF CONSTRUCTION

The Department of Commerce is required to formulate standards for solar energy systems (Public Act 605, 1979).

Contact Energy Extension Service
 Michigan Energy Administration
 Department of Commerce
 P.O. Box 30228
 Lansing, MI 48909
 (517) 373-6430

MINNESOTA

TAX INCENTIVES

The market value of solar, wind, or agriculturally derived methane gas systems used for heating, cooling, or electricity in a building or structure is excluded from property tax. The installations must be done prior to 1/1/84 (Chapter 786, Laws of 1978).

Contact Local assessor or board of assessors

This law provides an individual income tax credit of 20% of the first \$10,000 spent on renewable energy source equipment installed on a Minnesota building of six dwelling units or less. Eligible expenditures include: those eligible as federal renewable energy source property (solar, wind, and geothermal); earth-sheltered dwellings; equipment producing ethanol, methanol or methane for fuel, but not for resale; passive solar energy systems. Excess credit can be carried forward through 1984. Federal regulations of the U.S. Internal Revenue Service shall be used to administer relevant portions of the credit. Expenditures must be made between 1/1/79 and 12/31/82 (Chapter 303, Laws of 1979).

Contact Department of Revenue
Centennial Office Building
658 Cedar Street
St. Paul, MN 55145
(612) 296-3781

LAND USE

Zoning ordinances may provide for the protection of solar access for solar energy systems. Solar easements are recognized and the contents are prescribed; they are enforceable in civil actions. Depreciation resulting from easements (but not any appreciation) shall be included in revaluation for property tax (Chapter 786, Laws of 1978).

Contact (for zoning) Local planning or zoning body

Local governments are prohibited from preventing earth-sheltered construction as long as it otherwise complies with local zoning ordinances. Variances can be given to facilitate earth-sheltered construction. An appropriation of \$20,000 is made to conduct a study of impediments to earth-sheltered construction (Chapter 2, Special Session, Laws of 1979).

Contact Minnesota Energy Agency
980 American Center Building
150 E. Kellogg Boulevard
St. Paul, MN 55101
(612) 296-5120

STANDARDS AND REGULATION OF CONSTRUCTION

The Building Code Division of the Department of Administration is required to promulgate performance standards for solar energy systems (Chapter 333, Laws of 1976).

Contact Minnesota Department of Administration
Building Code Division
408 Metro Square Building
St. Paul, MN 55101
(612) 296-4639

MISSISSIPPI

TAX INCENTIVES

Labor, property or services used in the construction of solar energy heating, lighting, or electric generating facilities used by universities, colleges, or junior colleges are exempted from sales tax. Expires 1/1/83 (§27-65-105 of the Mississippi Code).

Contact Mississippi Tax Commission
Sales Tax Division
P.O. Box 960
Jackson, MS 39205
(601) 354-6274

MISSOURI

LAND USE

This law declares that the right to use solar energy is a property right, but it cannot be acquired by eminent domain. Solar easements are recognized and subjected to the same conveyancing and recording requirements as other easements. The contents are mandated (§442.021 of the Missouri Code).

MONTANA

TAX INCENTIVES

Energy systems using non-fossil fuel energy (such as solar, wind, solid wastes, decomposition of organic wastes, solid wood wastes, and small scale hydroelectric) installed in an income tax-payer's dwelling before 12/31/82 are eligible for an income tax credit of 10% of the first \$1000 and 5% of the next \$3000. If a federal tax credit is also claimed, the state credit is reduced to 5% of the first \$1000 and 2 1/2% of the next \$3000 (Chapter 548, Laws of 1975; Chapter 574, Laws of 1977; Chapter 652, Laws of 1979).

This law provides individual or corporate income tax deductions for energy conservation improvements, including storm windows and insulation. It applies to all types of buildings at the following rates:

Residential buildings	Non-residential buildings
100% of 1st \$1000	100% of 1st \$2000
50% of 2nd \$1000	50% of 2nd \$2000
20% of 3rd \$1000	20% of 3rd \$2000
10% of 4th \$1000	10% of 4th \$2000
	(Chapter 576, Laws of 1977).

Contact State Department of Revenue
Income Tax Section
Sam Mitchell Building
Helena, MT 59601
(406) 449-2837

This law provides a 10-year real estate tax exemption for capital investments in non-fossil forms of energy generation as defined in the state income tax law (Montana Code Annotated 15-32-102). The maximum exemptions are \$20,000 for a single-family dwelling and \$10,000 for other buildings (Chapter 639, Laws of 1979).

Contact State Department of Revenue
Property Assessment Division
Sam Mitchell Building
Helena, MT 59601
(406) 449-2808

GRANTS AND LOANS

This law permits utility companies to install and finance energy conservation materials and non-fossil forms of energy generation systems. The interest rate on energy conservation loans can be no less than two percentage points below the discount rate on 90-day commercial paper in the Ninth Federal Reserve District. The interest rate on non-fossil energy loans shall be 5% to 7% annually. Financial institutions may lend money for energy conservation materials and non-fossil energy systems at no less than two percentage points below the Federal Reserve rate. Any interest foregone by not charging the prevailing rate of interest for home improvement loans may be claimed as a credit against energy producer's license tax by a utility, or against corporation license tax by a financial institution (Montana Code Annotated 15-32-107).

Contact Local lending institution or utility company

This law amends the Montana Code (MCA90-4-101) to permit the Department of Natural Resources and Conservation to participate in commercial as well as non-commercial projects under its renewable resources research, development and demonstration program (Chapter 624, Laws of 1979).

Contact Energy Division
Department of Natural Resources and Conservation
Capitol Station
Helena, MT 59601
(406) 449-3940

LAND USE

This law recognizes solar easements and subjects them to the same conveyancing and recording requirements as other easements. The contents are prescribed (Chapter 524, Laws of 1979).

NEBRASKA

LAND USE

This law recognizes solar easements and prescribes their contents. Easements can be enforced in a civil suit. Local governments may include solar access considerations in their zoning ordinances and development plans. Variances from other ordinances may be granted to facilitate solar access (Legislative Bill 353, 1979).

Contact Local zoning board or planning commission

NEVADA

TAX INCENTIVES

This law establishes a property tax allowance on solar, wind, geothermal, water-powered, or solid waste energy systems in residential buildings. The property tax allowance equals the difference in tax on the property with the energy system and the tax on the property without the energy system. The allowance may not exceed the tax accrued or \$2000, whichever is less. Claims are to be filed with the county assessor (Chapter 345, Laws of 1977).

Contact Local county assessor

LAND USE

This law formally recognizes solar easements and prescribes their contents. The easement will run with the land upon transfer of title but can terminate upon expiration or release (Chapter 314, Laws of 1979).

STANDARDS AND REGULATION OF CONSTRUCTION

The Nevada Department of Energy is required to establish energy conservation standards, including provisions on the design and construction of solar, geothermal, wind or other renewable energy systems. Standards must then be included in all city and county building codes (Chapter 17, Laws of 1979).

Contact Nevada Department of Energy
1050 E. Williams
Suite 405
Capitol Complex
Carson City, NV 89710
(702) 885-5157

NEW HAMPSHIRE

TAX INCENTIVES

Cities and towns are enabled to grant property tax exemptions to property owners with solar heating, cooling, or hot water systems and will decide the amount of the exemption and the manner of determination. An application for the exemption must be filed with the local assessor (Chapter 391, Laws of 1975; Chapter 5202, Laws of 1977).

Contact Local assessor or board of assessors

NEW JERSEY

TAX INCENTIVES

Solar heating and cooling systems, including sea thermal gradients and wind-powered systems, are exempt from property tax. Systems must be certified under the State Uniform Construction Act on forms designated by the Division of Taxation. Systems must meet standards established by the State Department of Energy. Expires 12/31/82 (Chapter 256, Laws of 1977).

Contact Local assessor or board of assessors

Solar energy devices designed to provide heating, cooling, electrical or mechanical power are exempt from sales tax. These systems must meet the standards established by the state Department of Energy (Chapter 465, Laws of 1977).

Contact State Division of Taxation
Tax Counselors
P.O. Box 999
Trenton, NJ 08646
(609) 292-6400

LAND USE

Solar easements are recognized and subject to the same requirements as other easements; the contents are prescribed (Chapter 152, Laws of 1978).

STANDARDS AND REGULATION OF CONSTRUCTION

The Division of Energy Planning and Conservation of the State Department of Energy is required to adopt standards for solar energy systems (Chapter 256, Laws of 1977).

Contact Department of Energy
101 Commerce Street
Newark, NJ 07102
(201) 648-3290

NEW MEXICO

TAX INCENTIVES

This law provides for an income tax credit of 25% of the cost of a solar energy system or a maximum of \$1000. It is available for solar energy systems which heat or cool the taxpayer's residence and for swimming pool heating systems. The criteria of the Solar Heating and Cooling Demonstration Act of 1974 (42 USC 5506) must be met. Credit in excess of the taxes due will be refunded (Chapter 12, Laws of 1975; Chapter 170, Laws of 1978; Chapter 353, Laws of 1979).

Individuals may claim an income tax credit for a solar energy system used in an irrigation pumping system. The system design must be approved by the Energy Resources Board prior to installation, and it must result in a 75% reduction in the use of fossil fuel. This law is not applicable if federal credit were claimed or if credit were claimed for this equipment under other provisions of the state law. Credit in excess of the taxes due will be refunded (Chapter 114, Laws of 1977).

Contact State Department of Taxation & Revenue
Income Tax Division
P.O. Box 630
Santa Fe, NM 87503
(505) 827-3221

LAND USE

The right to use solar energy is a property right of landowners; disputes regarding access will be settled by rule of prior appropriation (Chapter 169, Laws of 1977).

STANDARDS AND REGULATION OF CONSTRUCTION

This law directs the New Mexico Solar Energy Research and Development Institute to develop performance standards for solar energy equipment (Chapter 347, Laws of 1977).

Contact New Mexico Solar Energy
Research and Development Institute
Box 3 SOL
New Mexico State University
Las Cruces, NM 88003
(505) 646-1846

NEW YORK

TAX INCENTIVES

This law provides a property tax reduction for owners of solar or wind energy systems. Passive systems qualify to a limited extent. The reduction of assessment is equal to the difference between assessment of the property with the energy system and assessment of the property without the system. The system must conform to guidelines of the state energy office and must be installed before 7/1/88. The exemption is good for 15 years after it is granted (Chapter 322, Laws of 1977; Chapter 220, Laws of 1979).

Contact Local assessor or board of assessors

LAND USE

This law recognizes solar easements and subjects them to the same conveyancing and recording requirements as other easements. The contents are prescribed (Chapter 705, Laws of 1979).

This law amends the general city law, the village law, and the town law to make the protection of solar access a valid purpose of zoning regulations. Effective 1/1/80. Before 9/30/80 the state energy office must issue guidelines to assist local governments in implementing the act (Chapter 742, Laws of 1979).

Contact Local zoning board or
New York State Energy Office
Agency Building 2
Empire State Plaza
Albany, NY 12223
(518) 474-8181

STANDARDS AND REGULATION OF CONSTRUCTION

This law requires that the Commissioner of the State Energy Office promulgate guidelines and definitions for solar energy systems (Chapter 322, Laws of 1977).

Contact New York State Energy Office
Agency Building 2
Empire State Plaza
Albany, NY 12223
(518) 474-8181

NORTH CAROLINA

TAX INCENTIVES

This law provides for a corporate and individual income tax credit of 25% of the cost of a solar heating, cooling, or hot water system. There is a maximum credit of \$1000 per unit or building. Although this credit may be taken only once, the amount of credit may be spread over 3 years. The system may be in any type of building, and it must meet the performance criteria of the U.S. Secretary of the Treasury or the North Carolina Secretary of Revenue (Chapter 792, Laws of 1977; Chapter 892, Laws of 1979).

Contact State Department of Revenue
 Income Tax Division
 P.O. Box 25000
 Raleigh, NC 27640
 (919) 733-3991

Buildings with solar heating or cooling systems shall be assessed as though they had a conventional system. Expires 12/1/85 (Chapter 965, Laws of 1977).

Contact Local assessor or board of assessors

NORTH DAKOTA

TAX INCENTIVES

This law provides for an income tax credit for solar or wind energy devices. The credit is 5% per year for 2 years. The system must provide heating, cooling, mechanical, or electrical power (Chapter 537, Laws of 1977).

Contact State Tax Commission
 Income Tax Division
 Capitol Building
 Bismarck, ND 58505
 (701) 224-3450

Solar heating or cooling systems in any building are exempt from property tax for 5 years after installation (Chapter 508, Laws of 1975).

Contact Local assessor or board of assessors

LAND USE

Solar easements are recognized and subject to the same requirements as other easements; the contents are prescribed (Chapter 425, Laws of 1977).

OHIO

TAX INCENTIVES

Solar, wind, and hydrothermal energy systems installed through 12/31/85 are exempt from real estate tax. A corporate franchise tax credit of 10% of the cost of a solar or wind energy system is created. The law creates a sales tax exemption for materials sold to a construction contractor for use in a solar, wind, or hydrothermal energy system. Sales of these systems are exempt from sales tax; components and labor costs are also covered by the exemption. This exemption is valid through 12/31/85.

A personal income tax credit of 10% of the cost of a solar, wind, or hydrothermal energy system is created. The system must be installed in a building that is owned and occupied as a dwelling, or owned and operated by the taxpayer in Ohio. The credit includes a two-year carry forward provision and a maximum credit of \$1000. To qualify for any of these tax advantages, the taxpayer's system must provide space heating or cooling, hot water, industrial process heat, mechanical or electrical energy and must meet guidelines established by the Department of Energy. Passive designs are included to a limited extent (Amended Substitute House Bill 154, 1979).

Contact (for income and franchise tax information)	Ohio Tax Commission Income Tax Division 1030 Freeway Drive Columbus, OH 43229 (614) 466-7910
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Contact (for guidelines)

Ohio Department of Energy
30 E. Broad Street, 34th Floor
Columbus, OH 43215
(614) 466-7915

Contact (for sales tax information)

Ohio Tax Commission
Sales Tax Division
30 W. Broad Street
Columbus, OH 43215
(614) 466-7350

LAND USE

This law recognizes solar easements and subjects them to the same requirements as other easements. The contents are prescribed (Amended Substitute House Bill 154, 1979).

OKLAHOMA**TAX INCENTIVES**

Individuals may claim an income tax credit for solar energy devices used to heat, cool, or furnish electrical or mechanical power at their principal residence. The credit is equal to 25% of the cost of the system or a maximum of \$2000. Although this credit may be taken only once, the amount of credit may be spread over 3 years. Expires on 1/1/88 (Chapter 209, Laws of 1977).

Contact State Tax Commission
 Income Tax Division
 2501 Lincoln Boulevard
 Oklahoma City, OK 73194
 (405) 521-3125

OREGON**TAX INCENTIVES**

This law exempts, in addition to solar systems, geothermal, wind, water and methane gas energy systems from real estate tax until 1/1/98. Property owned or leased by persons producing, transporting, or distributing energy is not included (Chapter 196, Laws of 1977; Chapter 670, Laws of 1979).

Contact Local assessor or board of assessors.

An income tax credit is provided for owners, tenants, and lessors of property used as a principal or secondary residence. The credit equals 25% of the cost of alternative energy devices using solar, water, wind, or geothermal resources for 10% of the energy requirements of the dwelling unit or 50% of the water heating requirements of the unit. Maximum credit is \$1000 per dwelling unit. A carry forward provision is included. Systems must be certified by the Department of Energy before a credit may be claimed (Chapter 196, Laws of 1977; Chapter 670, Laws of 1979).

This law provides an income tax credit for weatherization materials installed in the taxpayer's principal residence, including a mobile or floating home or a unit of a multifamily dwelling. A list of qualifying items is available from the Department of Energy or the Department of Revenue. Items must be installed before 1/1/85. The credit equals 25% of the cost of eligible items to a maximum of \$125. A 5-year carry forward provision is included (Chapter 534, Laws of 1979).

This law provides a corporation excise tax credit for commercial lending institutions making loans at 6 1/2% interest or less for the installation of certified alternative energy devices. Maximum loans are \$10,000 for home improvement loans. The tax credit is equal to the difference between 6 1/2% and the average interest rate for home improvement loans made in a previous calendar year (Chapter 483, Laws of 1979).

This law creates a corporate income tax credit of 35% of the cost of energy conservation facilities. The credit is claimed at a rate of 10% for the first two years and 5% for each of the next three years. A carry forward provision is included.

Energy conservation facilities means facilities used in trade or business and employing or processing renewable energy sources to: (1) replace a substantial part of an existing use of electricity, petroleum, or natural gas; (2) provide initial use of energy where such resources would have been used; (3) generate electricity to replace an existing source of electricity or provide a new source of electricity; or (4) perform a process that obtains energy resources from material that would otherwise be solid waste. Renewable energy resources include, but are not limited to straw, forest slash, wood waste or other forms of forest waste, industrial or municipal waste, solar energy, wind power, water power or geothermal energy (Chapter 512, Laws of 1979).

Contact State Department of Revenue
 State Office Building
 Salem, OR 97310
 (503) 378-3366

GRANTS AND LOANS

This law creates a loan fund for alternate energy projects and authorizes the Director of the Department of Energy to sell bonds to finance the loan fund (Chapter 732, Laws of 1977).

Contact Oregon Department of Energy
 Room 111
 Labor & Industries Building
 Salem, OR 97310
 (503) 378-4128

To finance a domestic solar energy system, veterans can obtain a loan in excess of the maximum allowed under the War Veterans Fund. The system must provide at least 10% of the home's energy requirements and must meet performance criteria established by the State Department of Energy (Chapter 315, Laws of 1977).

Contact State Department of Veterans' Affairs
 3000 Market Street Plaza
 Suite 522
 Salem, OR 97310
 (503) 378-6438

LAND USE

This law enables local governments to regulate solar access in comprehensive plans, zoning ordinances, and subdivision regulations. Solar easements are recognized and their contents are prescribed. Private restrictions prohibiting the use of solar energy are void and unenforceable if the provision is executed after 10/3/79 (Chapter 671, Laws of 1979).

Contact Local zoning board or planning commission

STANDARDS AND REGULATION OF CONSTRUCTION

The Department of Energy is required to adopt rules prescribing performance criteria for solar energy systems (Chapter 196, Laws of 1977).

Contact Oregon Department of Energy
 Room 111
 Labor & Industries Building
 Salem, OR 97310
 (503) 378-4128

RHODE ISLAND

TAX INCENTIVES

Solar heating or cooling systems in residential or non-residential buildings shall be assessed at no more than the value of a conventional system necessary to serve the building. Law expires 4/1/97 (Chapter 202, Laws of 1977).

Contact Local assessor or board of assessors

SOUTH DAKOTA

TAX INCENTIVES

This law provides property tax assessment credit for renewable resource energy systems (solar, wind, geothermal, and biomass). For residential property the amount of the credit equals the assessed value of the property with the system, minus the assessed value of the property without the system, but not less than the actual installation cost of the system. The credit for systems in commercial buildings is equal to 50% of the cost of installation. For residential buildings, full credit is given for 5 years. For the next 3 years, the credit is 75%, 50%, and 25% of the full credit. For commercial buildings, full credit is given for 3 years, and for the next 3 years credit is 75%, 50%, and 25% of the full credit. Taxpayers must apply to the county auditor (Chapter 74, Laws of 1978).

Contact Local county auditor

TENNESSEE

TAX INCENTIVES

Solar or wind energy systems for heating, cooling, or electrical power shall be exempt from property taxation. Law expires 1/1/88 (Chapter 837, Laws of 1978).

Contact Local assessor or board of assessors

GRANTS AND LOANS

Provides loans to low- and moderate-income persons to make energy conserving improvements, including the installation of solar hot water systems (Chapter 884, Laws of 1978).

Contact Tennessee Housing Development Authority
Hamilton Bank Building
Nashville, TN 37219
(615) 741-3023

LAND USE

This law recognizes solar easements and prescribes their contents. They are subjected to the same general requirements as other easements. The Tennessee Energy Authority is directed to prepare a sample solar easement for use in Tennessee. Local governments are empowered to protect solar access through zoning regulations (Chapter 259, Laws of 1979).

Contact (for sample easement)	Tennessee Energy Authority Suite 707 Capitol Boulevard Building Nashville, TN 37219 (615) 741-2994
Contact (for zoning)	Local zoning or planning body

TEXAS

TAX INCENTIVES

The legislature is allowed to exempt solar- or wind-powered energy devices from property tax. (Chapter 719, Laws of 1975). (Article VIII, Sec. 2(a) of Texas Constitution).

This law exempts solar and wind energy devices from real estate tax assessments. The devices must be used for thermal, mechanical or electrical energy. The Comptroller of Public Accounts shall develop guidelines to assist tax assessors in carrying out this law. Effective 1/1/80 (Chapter 107, Laws of 1979).

This law provides a franchise tax exemption for corporations exclusively engaged in manufacturing, selling, or installing solar energy devices for heating, cooling, or electrical power (Chapter 584, Laws of 1977).

Solar energy systems used for heating, cooling, or electrical power are exempt from sales tax. Corporations may deduct from taxable capital the amortized cost of a solar energy device over a period of 60 months or more.

Contact Comptroller of Public Accounts
 Capitol Station
 Drawer SS
 Austin, TX 78775
 (512) 475-2206

UTAH

LAND USE

This law recognizes solar easements as a property interest. Easements must be in writing and they will run with the land in perpetuity unless terminated upon stated conditions. Enforcement may be by injunction or other civil action (Chapter 82, Laws of 1979).

VERMONT

TAX INCENTIVES

Towns may enact a property tax exemption for alternate energy systems. Systems exempted are grist mills, windmills, solar energy systems, and devices to convert organic matter to methane. All components are exempt, including land on which the facility is situated, up to one-half acre (Act 226, 1976).

Contact Local assessor or board of assessors

Wood-fired central heating and solar or wind systems for heating, cooling, or electrical power are eligible for income tax credit if they are installed in the taxpayer's dwelling before 7/1/83. The credit is equal to 25% of the cost of the system or \$1000, whichever is less. Businesses may deduct 25% of the cost of the system or \$3000, whichever is less (Act 210, 1978).

Contact State Tax Department
 Income Tax Division
 State Street
 Montpelier, VT 05602
 (802) 828-2517

VIRGINIA

TAX INCENTIVES

Any county, city, or town may exempt solar energy equipment used for heating, cooling, or other applications from property tax. The State Board of Housing must certify the system. The exemption is good for not less than 5 years (Chapter 561, Laws of 1977).

This law creates a separate class of tangible personal property for local taxation. The class includes energy conversion equipment purchased by a manufacturer for the purpose of changing the energy source of a plant from oil or gas to coal, wood, or alternative energy resources. Cogeneration equipment is also included. Tax years covered are those beginning after 7/1/79. This class of property may be taxed at a rate different from the rate on other tangible personal

property, but not higher than the rate on machinery and tools. To be eligible, equipment must have been purchased after 12/31/74 (Chapter 351, Laws of 1979).

Contact Local taxing authority

LAND USE

This law subjects solar easements to the same legal requirements as other easements and mandates contents of the agreement (Chapter 323, Laws of 1978).

WASHINGTON

TAX INCENTIVES

Solar water and space heating or solar power systems are exempt from property taxation. Claims must be filed with the county assessor. The exemption is valid for 7 years. Claims must be filed by 12/31/81 (Chapter 364, Laws of 1977).

Contact Local county assessor

GRANTS AND LOANS

This law authorizes municipally or privately owned utility companies to establish programs to perform energy audits, to recommend improvements and to arrange the installation and financing of energy conservation materials in residential buildings. (Chapter 239, Laws of 1979).

Contact Local utility company

LAND USE

This law permits local governments to regulate protection of solar access in comprehensive plans and zoning ordinances. It recognizes easements, covenants and other restrictions on the use of real property, created to protect access to sunlight. The contents of easements are mandated and they are subjected to the same conveyancing and recording requirements as other easements. Some remedies for interference with a solar easement are authorized (Chapter 170E-1, Laws of 1979).

Contact Local planning commission or zoning board

WISCONSIN

TAX INCENTIVES

The cost of alternative energy systems owned and installed by corporations on property in Wisconsin between 4/20/77 and 12/31/84 may be used as a tax deduction in the year paid for, may be depreciated, or may be amortized over 5 years. This law covers solar, waste conversion and wind systems certified by the Department of Industry, Labor and Human Relations (Chapter 313, Laws of 1977).

Contact State Department of Revenue
Income Tax Division
P.O. Box 8910
Madison, WI 53708
(608) 266-1911

GRANTS AND LOANS

This law creates a program to subsidize the cost of an alternative energy system purchased by individuals. If the building on which the system is installed was on the local tax rolls before 4/20/77, the rate of refund will be: 24% in 1979 and 1980; 18% in 1981 and 1982; 12% in 1983 and 1984. Eligible expenses must exceed \$500, but the subsidy will not be calculated on more than \$10,000. Eligible systems are: 1) solar systems used for space heating or cooling, crop drying, electricity or water heating; 2) waste conversion energy systems including equipment which converts waste into usable forms of energy, but excluding solid-fuel consuming devices used for residential purposes; and 3) wind energy systems that convert wind energy into other usable forms of energy. All systems must meet standards of the Department of Labor, Industry and Human Relations (Chapter 34, Laws of 1979).

Contact Department of Industry, Labor & Human Relations
 201 E. Washington
 Rm. 101, Safety and Buildings Division
 Madison, WI 53702
 (608) 266-1149

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 13

OTHER ECONOMIC CONSIDERATIONS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO



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OBJECTIVES

The objectives in this module are to:

1. Determine the cost of solar thermal energy delivered from a system.
2. Establish cost effectiveness of a system.
3. Determine an economic optimum collector area for a system.

INTRODUCTION

In a preceding module, two procedures are described for calculating the total life-cycle cost (TLCC) of solar heating systems and the TLCC of conventional heating systems. If the TLCC of the solar system is less than the TLCC of a conventional system, the solar system is considered to be cost effective. The first procedure described yields the TLCC of both solar and conventional systems directly but assumes constant interest, inflation, and discount rates during the period of analysis. The second procedure, although more tedious in calculation, allows TLCC determinations with variable rates of interest, inflation, and discount in any year according to an economic "scenario". In both methods, discount, inflation, and interest rates can be treated in "real" terms, or alternatively, discount and inflation rates can be treated in marginal terms relative to the general inflation rate for goods and services.

The concept of life-cycle costs applied to residential systems is probably too complex for the average homeowner; also, determination of cost effectiveness is susceptible to assumed energy inflation and

discount rates. Therefore, there may be some utility in appraising the economic value of solar systems by alternative approaches presented in this module.

BREAKEVEN COST

Instead of attempting to predict the rates of inflation explicitly in the analysis, a simple approach is a calculation of the breakeven cost of the solar system. Breakeven occurs when the cost of energy delivered by the solar system is equal to the cost of the conventional energy it displaces and is usually calculated in terms of a uniform annual cost. In simplest terms, inflation and discount rates are ignored, and the uniform annual cost of the solar system is considered to consist of annual mortgage repayment (principal plus interest) and operating costs. More complex breakeven analysis would include other annual costs such as property taxes, insurance, and maintenance minus an annual credit for income tax saving on interest paid on the mortgage. The simple calculation assumes that property taxes, insurance, and maintenance costs are offset by an annual income tax credit on the interest paid (at least in the first few years). A breakeven cost calculation is illustrated by the following example.

EXAMPLE 13-1

A solar heating system with 400 ft² of collectors costs \$14,000 (\$35/ft²). After taking a Federal income tax credit of \$4,000 and a State income tax credit of \$1,000, a 30-yr loan is negotiated for the balance of \$9,000 at 10 percent interest rate. The solar system will

provide an annual average useful thermal energy of 65 million Btu. Determine a (simple) breakeven cost for the solar system.

SOLUTION

At 10 percent interest, a 30-yr loan will necessitate a uniform annual payment of \$955 to repay the \$9,000 loan. Since the average net annual energy delivery is \$65 MMBtu, the solar energy cost, or breakeven cost, is \$14.69/MMBtu. In terms of various types of conventional energy used for heating, the breakeven prices of the conventional fuels are:

Type of Furnace	Breakeven Price
Electrical resistance	5.0¢/kWh
Oil furnace at 60 percent efficiency (140,000 Btu/gal)	\$1.23/gal
Propane furnace at 70 percent efficiency (90,000 Btu/gal)	\$0.93/gal
Natural gas furnace at 70 percent efficiency (1,000 Btu/ft ³)	\$1.03/100ft ³

PAYBACK

Another economic measure for a solar system is the number of years (elapsed time) required for accumulated energy cost savings to equal initial investment (payback period). The calculation may be made with or without consideration to escalation of energy prices and market discount rate. If inflation and discount rates are ignored, the result of the calculation is called simple payback (SPB) period. If inflation and discount are both considered, the result is called discounted

payback (DPB) period. In equation form, simple payback period may be determined as follows:

$$\text{SPB(yrs)} = \frac{\text{System Cost}}{(\text{Annual Energy Cost Savings} - \text{Annual O and M Costs})} \quad (13-1)$$

Discounted payback period is the period between initial investment and the time when the sum of net energy savings, appropriately inflated and discounted, equals the initial investment. A nomograph to determine both simple payback and discounted payback periods is provided in Figure 13-1. Its use is illustrated in Example 13-2.

EXAMPLE 13-2

Determine the simple payback period for a solar system with a net investment cost (after tax credits) of \$8,000 which delivers 80 MMBtu of useful solar energy annually. The annual operating and maintenance costs total \$172, and the conventional fuel displaced is electricity at 5¢/kWh. Determine also the discounted payback period if the fuel inflation rate is 15 percent and discount rate is 10 percent.

SOLUTION

The annual energy cost savings is

$$\frac{80 \times 10^6 \text{ Btu}}{3423 \text{ Btu/kWh}} \times \frac{\$0.05}{\text{kWh}} = \$1172 .$$

The operating cost is computed on the basis of 7 percent operating energy requirement and a small amount is added for maintenance to total \$172. Then the simple payback period is

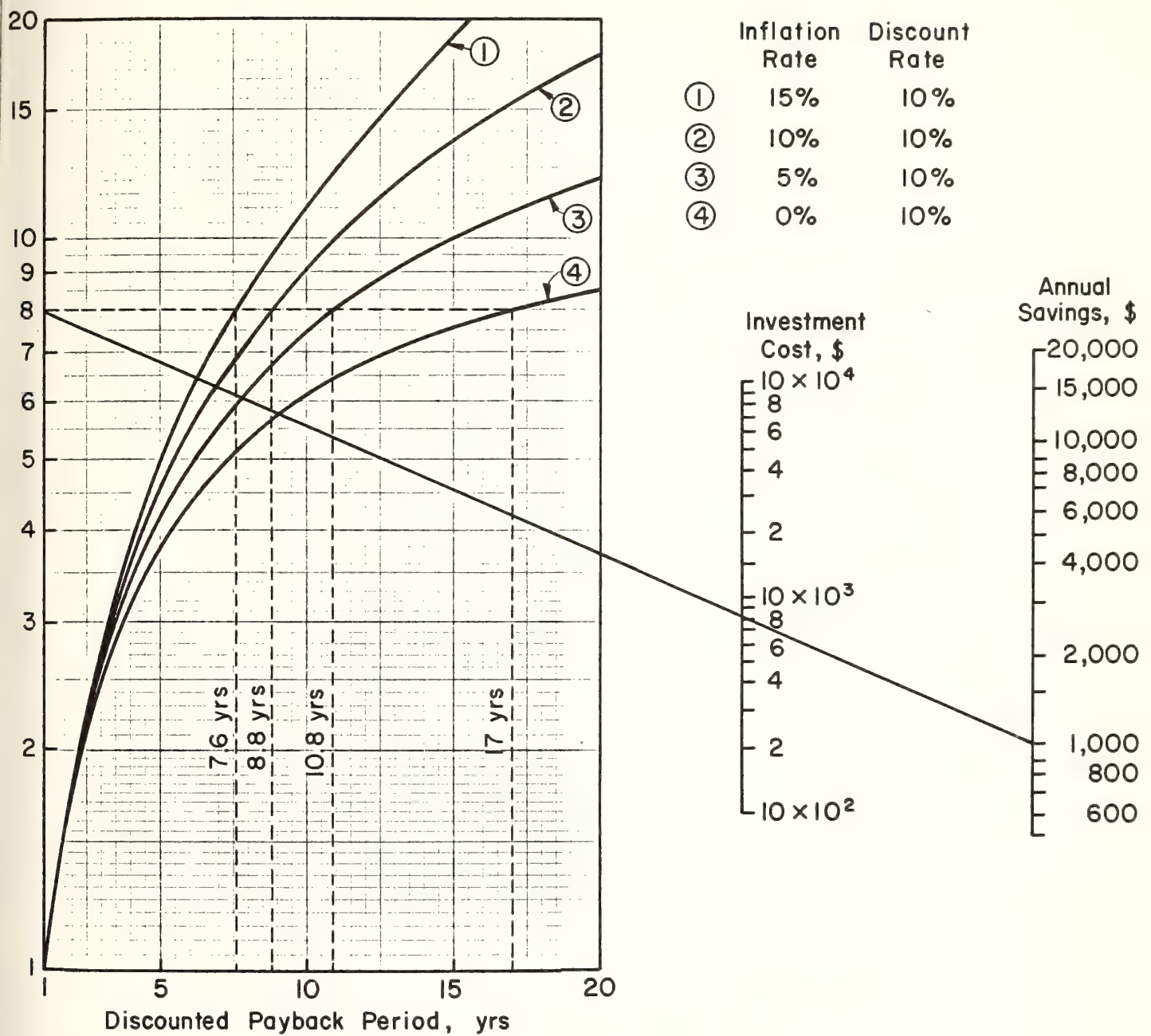


Figure 13-1. Nomograph for Determining SPB and DPB

$$SPB = \frac{80000}{1172-172} = 8 \text{ years}$$

Using the nomograph of Figure 13-1, the solid line connecting \$1000 annual savings with \$8000 investment cost leads to a simple payback of 8 years as calculated above. Also, following the dashed horizontal line to curve (1), the discounted payback period is 7.6 years. If the fuel inflation rate and discount rate are both 10 percent, the payback period is slightly longer, as indicated by the intercept of the horizontal dashed line and curve (2). Other combinations of inflation rates and 10 percent discount rate may be read from the nomograph by interpolating between the appropriate curves.

ECONOMIC OPTIMUM COLLECTOR AREA

There is, in most situations, an economically optimum collector area in the sense that annual heating cost with a combined solar and conventional energy system is minimum for a specific solar system size. However, optimum area, (A^*), is dependent upon the economic assumptions made in the analysis. Using the life-cycle cost approach, the concept of optimum collector area is illustrated in Figure 13-2.

The curve labeled LCC_s is the life-cycle cost for the solar system alone, LCC_c is the life-cycle cost for conventional energy required as auxiliary to the heating system. The sum of LCC_s and LCC_c is the total life-cycle cost for the system, TLCC. As collector area increases, LCC_s increases (nearly linearly with collector area) and LCC_c decreases because solar energy supplies an increasing fraction of the total heat requirements. Depending upon the shape of the two curves, there will

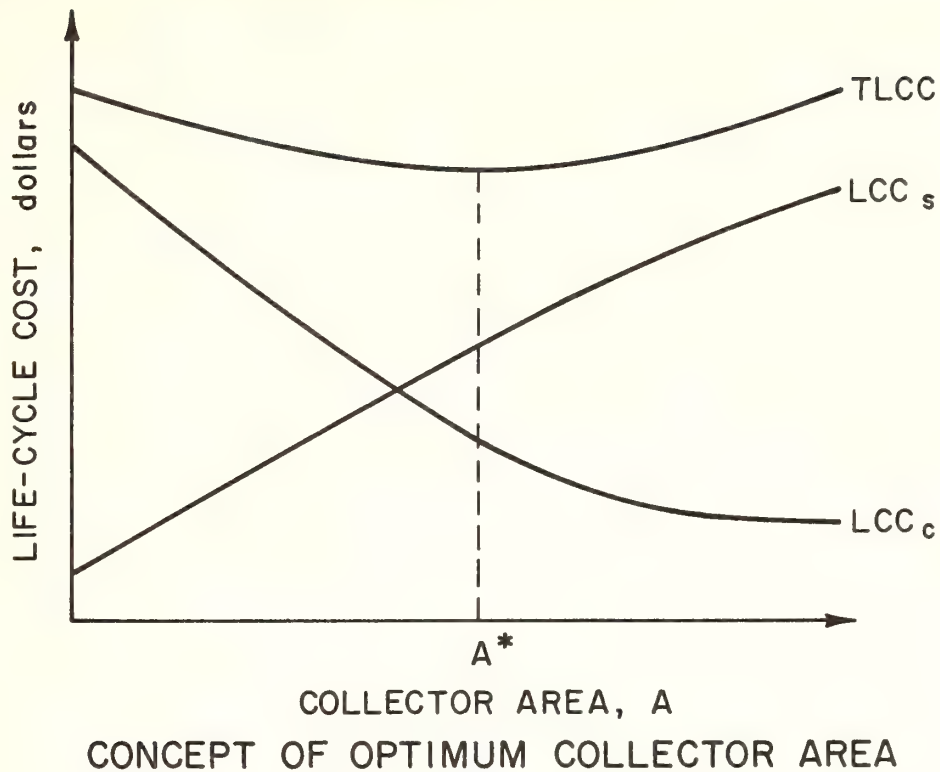


Figure 13-2. Concept of Optimum Collector Area

generally be a minimum point in the TLCC curve, and the collector area corresponding to the minimum is labeled the optimum collector area, A^* .

The determination of A^* is readily made by using the first procedure for computing TLCC described in the previous module and a simple equation-based method for determining the solar fraction such as the relative areas method described in Module 6. The TLCC of the solar/auxiliary system is given by Equation (12-3) and is repeated below:

$$TLCC = C_T(\text{solar}) = (AC_a)E_1 + C_oE_o + C_mE_m + (1-F)Lc_fE_f, \quad (13-2)$$

The minimum point in the TLCC curve occurs where

$$C_aE_1 - Lc_fE_f \frac{dF}{dA} = 0 \quad (13-3)$$

The functional relationship of annual solar fraction F and collector area A is expressed in the relative areas method (Module 6) as

$$F = c_1 + c_2 \ln\left(\frac{A}{A_0}\right) \quad (13-4)$$

Thus,

$$\frac{dF}{dA} = \frac{c_2}{A}, \quad (13-5)$$

and by substituting into Equation (13-3)

$$A^* = \frac{c_2 L c_f E_f}{C_a E_1} \quad (13-6)$$

where

- L is the total building heating load (MMBtu)
- c_f is cost of conventional energy (cost/MMBtu)
- c_2 is a location dependent constant (dimensionless)
- E_f is the economic factor for fuel cost
- E_1 is the economic factor for the system
- C_a is unit cost of the solar system (cost/ft²)

The use of Equation (13-6) is illustrated by the following example.

EXAMPLE 13-3

Determine the optimum collector area for a liquid-heating solar system on a building located in Washington, D.C., and calculate the annual solar fraction the optimum system will provide: The following information is given:

Solar System

1. Liquid-heating collectors: $F_R(\tau\alpha)_n = 0.75$

$$F_R U_L = 1.00 \text{ Btu}/(\text{hr} \cdot \text{ft}^2 \cdot ^\circ\text{F})$$

2. Building UA = 450 Btu/(hr·°F)
3. Auxiliary Heating - Electrical Energy

Economic Parameters

1. Electrical energy cost, $c_f = 5\text{¢/kWh} = \$14.65/\text{MMBtu}$
2. System cost, $C_a = \$28/\text{ft}^{2*}$
3. Mortgage terms: downpayment, $\alpha = 0.20$
period of loan, $m = 25$ yrs
interest rate, $i = 12$ percent
4. Property tax rate, $p = 0.02$
5. Insurance premium rate, $h = 0.003$
6. General inflation rate, $g = 0.06$
7. Fuel escalation rate, $r_f = 0.12$
8. Market discount rate, $d = 0.10$
9. Owner's income tax rate, $t = 0.35$
10. Life-cycle analysis period, $n = 20$ years

SOLUTION

From Module 12,

$$E_f = P/X(d, r_f, n) = 21.693$$

$$E_1 = \alpha + [(1-t)p+h]P/x(d,g,n) + (1-\alpha)[(1-t) \frac{P/x(d,o,m)}{P/x(i,o,m)} + (t) \frac{P/x(d,i,m)}{P/x(o,i,m)}]$$

$$E_1 = 0.2 + [(1-0.35)(0.02) + 0.003]13.082 + (1-0.2)[(1-0.35) \frac{9.077}{7.843} + (0.35) \frac{22.727}{98.347}]$$

*System cost consists of \$1000 fixed costs plus incremental costs of \$28/ft².

$$E_1 = 1.076$$

$$c_f = \$14.65/\text{MMBtu}$$

$$L = (UA)_{\text{bldg}} \times 24 \frac{\text{hr}}{\text{day}} \times DD$$

$$L = 450 \times 24 \times 4224 = 45.62 \text{ MMBtu}$$

$$c_2 = 0.271 \text{ (from Module 6)}$$

$$A^* = \frac{0.271 \times 45.62 \times 14.65 \times 21.693}{28 \times 1.076}$$

$$A^* = \underline{\underline{130 \text{ ft}^2 \text{ of collectors}}}$$

Using the relative areas method the solar fraction is calculated to be:

$$A_o = \frac{0.237(450)}{.75 - 1.0(.307)} = 241 \text{ ft}^2$$

$$F = .520 + .271 \ln(130/241) = \underline{\underline{0.35}}$$

REFERENCES

1. Barley, C.D. and Winn, C.B. (1978) "Optimal Sizing of Solar Collectors by the Method of Relative Areas", Solar Energy, Vol. 21, No. 4, pp. 279-289.
2. Barley, C.D. (1979) "Load Optimization in Solar Space Heating Systems," Solar Energy, Vol. 23, No. 2, pp. 149-156.
3. Brandemuehl, M.J. and Beckman, W.A. (1979), "Economic Evaluation and Optimization of Solar Heating Systems", Solar Energy, Vol. 23, No. 1, pp. 1-10.
4. Ruegg, R.T., McConnaughey, Sav, G.T., and Hockenbery, K.A. (1978), Life Cycle Costing. ABS Building Science Series 113, U.S. Dept. of Commerce, September.



TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 14

INSTALLATION OF SOLAR SYSTEMS

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO



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OBJECTIVE

The objective of this module is to explain preferred installation practices for solar systems. At the end of this module the trainee should be able to:

1. Identify a logical sequence for installing solar systems for new and retrofit construction.
2. Recognize items of specific concern for retrofit installations.

INTRODUCTION

The cost of labor to install a solar system is a substantial portion of the total solar system cost. Careful planning of the installation can significantly lower costs and improve the economic viability of a solar space conditioning and domestic water heating system. There are many important considerations in the installation of a system which affect not only initial costs, but operating, maintenance, and repair costs as well, and the latter factors can, in the long term, be as important as the first cost if the systems are not properly installed.

Whether the system is being installed in a new structure or is being retrofitted in an existing building, there are many common concerns. Other factors apply specifically to retrofit installations which may limit the options for locating storage tanks and other equipment inside the building, or mounting collectors on the roof. Scheduling the installation of the several parts of a solar heating system, whether for new or retrofit construction, is an important part of overall planning. An awareness of major factors when designing and planning system layouts

should result in minimum expense for installation, optimum operation, and low maintenance costs.

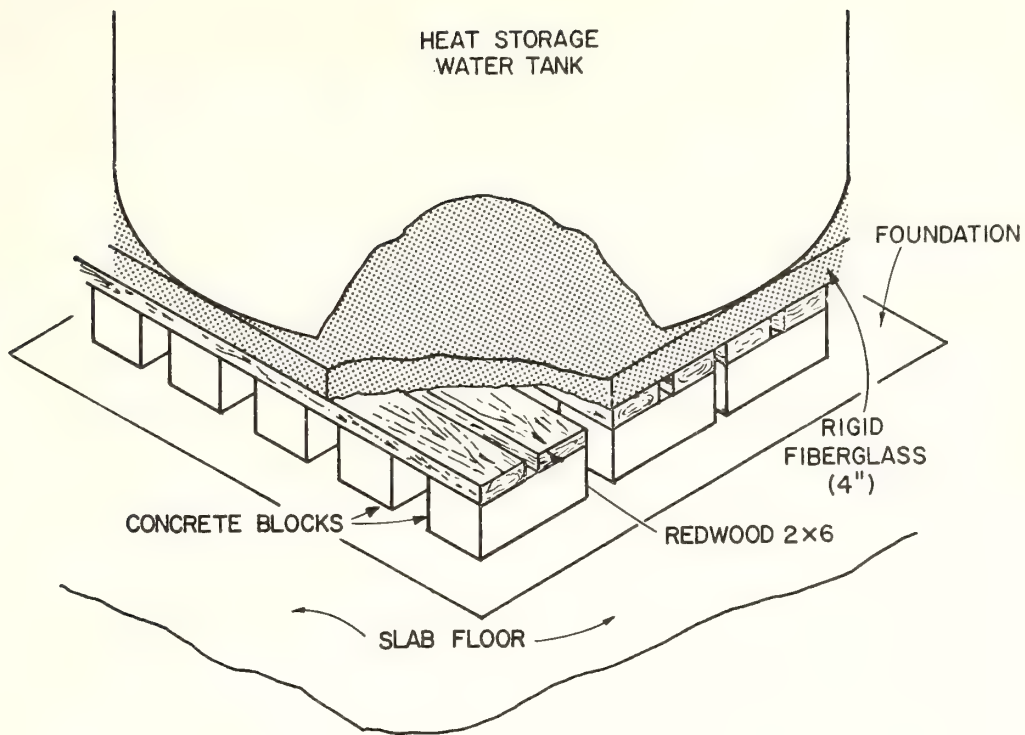
INSTALLATION OF STORAGE VESSELS

The first component of a solar system which should be installed in a new building is the solar storage vessel. The important factors of concern are foundation and footings, scheduling construction of the storage unit, placement of prefabricated tanks, anchoring the vessel, leak testing, insulating, and placing control sensors in the vessel. Many of these items are common to water tanks and pebble beds, and new and retrofit situations. However there are some important differences between water tanks and pebble beds that affect installation methods.

WATER TANKS

Several types of water containers are suitable for storage of solar-heated water. Regardless of the type of vessel used, the foundation must be adequate to support the weight of the tank when filled with water. If a cylindrical tank is placed vertically, the weight will be supported by the total area of the foundation. A typical arrangement is shown in Figure 14-1.

The weight of water is 62.4 lb/ft^3 , so a tank which is 6 ft high filled with water has a water load of about 375 lb/ft^2 . With an additional weight of about 300 lb for the steel tank, the total load on the foundation is about 390 lb/ft^2 . A concrete foundation at least 6 inches thick, reinforced with a 6-in. heavy wire mesh, is recommended to support the load. The type of soil beneath the foundation should also



BOTTOM INSULATION AND SUPPORT SCHEME FOR
WATER STORAGE TANK

Figure 14-1. Support for Vertical Water Storage Tanks

be considered. Sandy soil or gravel will cause no settlement problems, but if the soil contains a large amount of clay, expansion will take place when the soil becomes moist, and lift the foundation despite the weight of the water tank. In such cases, there should be a control joint in the concrete around the tank separating the foundation from the basement floor to prevent uncontrolled cracking of the foundation caused either by soil settlement or expansion.

Cylindrical water tanks are sometimes placed horizontally and supported by two or more saddle footings which rest on the foundation. The load of the water tank is concentrated at the saddles and the unit load will be larger than the load under a vertical tank. The foundation should be poured for the projected area of the tank so that the load on individual footings will be spread over the entire concrete pad.

It is recommended that storage tanks not be buried beneath the basement or garage because access for replacement or maintenance is difficult and expensive. Underground tanks outside the building meet with less objection, but tanks should preferably be placed above grade for accessibility. If there is no choice for placement except underground, the insulation on the tank must be completely protected from moisture penetration. There should also be provision for drainage in case of leakage from the tanks or from surface or ground water. Because of these factors, unless the tank can be placed inside the building, a separate underground chamber to house the storage tank is recommended.

All storage tanks should be inspected carefully for damage before and after placement. Although inspection may be difficult if a pre-insulated tank is purchased, pre-installation inspection may detect flaws so that unnecessary and costly replacement after installation can be avoided.

There should be at least 10 inches of fiberglass insulation, or equivalent, around a water tank to prevent excessive heat losses. It is desirable to achieve at least an R-30 insulation value. While heat losses from tanks placed within the heated envelope are distributed into the building and are utilized, these losses are uncontrolled and may contribute to overheating portions of the building, which is wasteful use of solar heat.

All tanks should be leak-tested after the piping is installed and connections are made and before insulation is applied. If the solar system is unpressurized, leak detection by filling the storage tank is satisfactory. However, if the system is to be pressurized the tanks should be pressure-tested.

The control sensor(s) should be placed in the tank before insulation is applied to the outside of the tank. An immersion well is suitable for thermistors, whether vertical or horizontal tanks are used.

PEBBLE BEDS

The foundation for rock storage boxes must be able to support a load of 500 to 600 lb/ft² depending upon the size and depth of pebbles used. Pebbles of 0.75- to 1.5-in. sizes weigh approximately 100 to 120 lb/ft³. In retrofit applications, constructing a container for a pebble bed is less difficult than installation of a water tank. However, placement of gravel into the box may pose some difficulties. In new construction, the pebbles should be placed before the subfloor is laid above the rock box. Considerable expense for gravel placement can be spared if the rocks can be loaded into the box directly from above rather than with a conveyor after the flooring has been placed.

The control sensor at the bottom of storage can be attached to the supporting screen after gravel has been placed, by reaching through the bottom port. In many systems, this sensor is mounted in the duct supplying cold air from storage and from the rooms to the collector. If a control sensor is needed at the top of the rock bed, likewise, the sensor can be inserted through the top port.

It is important that rock containers be air tight. All corners and joints in the box should be sealed from the inside with a pliable sealant. Many suitable sealant materials are available. Because sealing a pebble bed container from the outside is difficult, careful sealing during initial construction is very important.

Rock containers should be insulated at least to R-13. Less insulation is required for pebble-bed containers than for water tanks because the rate of heat conduction from the pebbles to the walls is low, and most of the heat lost through the walls comes from air circulating along the walls inside the storage container. Nevertheless, because the wall area of a container is large, insulation is needed to minimize these heat losses.

Locating storage in an existing building may be difficult without sacrificing valuable living space. Storage boxes for rocks can be located in the backyard above grade if desired. An underground rock bed may also be constructed, but the exterior should be sealed from moisture and the walls should be insulated to minimize heat losses. By far the most desirable location for a pebble bed is inside the heated space.

INSTALLATION OF COLLECTORS

The cost of installing collectors varies widely with design. Some manufacturers design collectors to minimize installation time and cost, while other manufacturers give little or no consideration to the installation process; although their collector costs may be low, the installation costs may be very high. Some manufacturers of liquid collectors provide plumbing fittings on the collector frame which are sturdy enough to be torqued with a pipe wrench. Other manufacturers simply provide a pipe stub protruding loosely from the frame and leave the problem of making piping connections to the installer. While collectors with prefitted connections may be more expensive than those without connections, the total installed cost of the former may be

considerably less than the latter. Similarly, collectors with provisions for anchoring the modules to the roof and for surface and edge flashing designed to fit the collectors can result in considerable time saving during installation. Air collectors with internal manifolding can result in considerable installation cost reduction compared to collectors that require two roof penetrations and separate ducting for each collector module.

Collectors are normally mounted on the roof of a building to minimize overall labor and material costs. In new construction, roof trusses should be angled at a suitable pitch, and collectors can be mounted either directly on the sheathing or integrally with the roof trusses. For retrofit construction, the pitch of the roof may not be at the desired angle, and collector tilt may have to be adjusted with standoffs above the finished roofing. If standoffs are used, a minimum spacing between the collector and the finished roofing is usually required by building codes. The spacing required will vary with location. Costs for mounting collectors with standoffs may be considerably higher than costs for mounting collectors directly on the roof. Wind and snow loads for standoff mounting will be larger than normal and the structural adequacy of roof trusses should be checked.

Rack-mounted collectors are suitable for flat roofs and placement on the ground. For ground-level installations, safeguards such as fences must be provided around the collector. An advantage of a rack-mounted array is that collectors can be assembled at ground level. A disadvantage, especially of the air types, is that duct runs and connections may be expensive.

PLUMBING AND DUCTING

After installing the collectors, pipe and duct connections are required. Headers must be connected to liquid collectors and ducts from manifolds to air collectors. Depending upon the type of collectors and arrangement of the array, headers and ducts may be connected to each collector or groups of collectors. For domestic water heating, requiring only a few collectors, headers may be connected to each module; but for space heating systems, collectors are usually interconnected in a two- or three-high array. In such arrangements, interconnection of collectors must be made during assembly.

If the absorber metal and piping in liquid collectors are of the same material, connections from the header to the collector may be made as illustrated in Figure 14-2. The "bent tube" arrangement allows thermal expansion and contraction of the header without creating excessive stress at the connectors. If, however, the absorber and pipe are dissimilar metals, a dielectric (non-metallic) coupling must be used. In that case, the offset between header and collector connections should be less than shown in Figure 14-2 to avoid crimping the flexible hose connections.

Headers should be sloped to drain. For closed loop systems, a slope of $1/4$ in. for each 10 feet of header is adequate, but for drain-down systems, a slope of 2 in. for each 10 feet is desirable. Headers should be covered with a hood, the base of which is flashed and the top of which should be easily removable for access to the header connections.

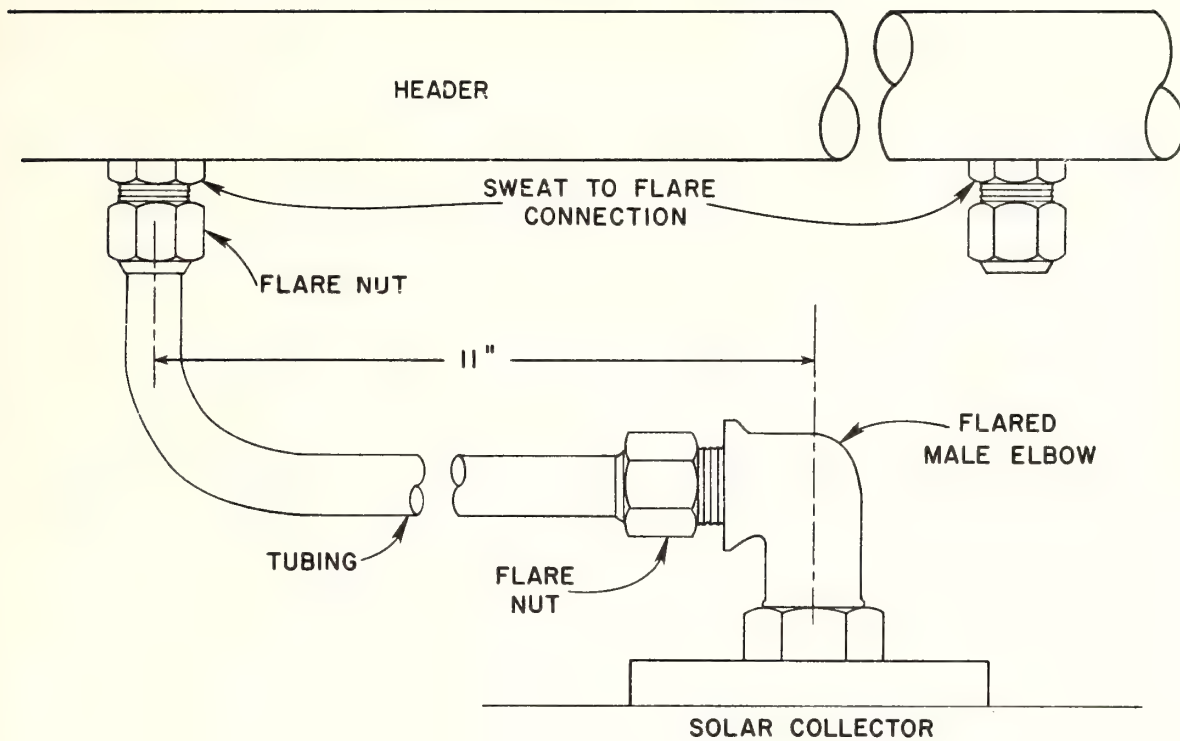


Figure 14-2. Connecting Collector to Header

Copper and high temperature plastic (CPVC) pipe are commonly used in solar installations. Of the two, copper is preferable, but if CPVC piping is used, supports must be provided at frequent intervals along horizontal runs so that the pipe will not sag when heated by a warm liquid. Long straight lengths of plastic piping should be avoided wherever possible. All piping should be leak-tested with water and pressurized to specified limits. Pipes in the collector loop, including the manifold, should be insulated with at least 1 inch of closed cell material.

It is extremely important that ducts in air systems be leak-tight, because performance can otherwise be significantly affected. Unlike in liquid systems, leaks are not easily detected, but if warm air escapes from a collector loop, cold air must enter the system elsewhere to make up for the air leak. Because leaks are difficult to avoid in a complex arrangement of collector arrays, the best practice is to position the blower in the loop so that air leakage is into the collector array. This infiltration into the collector provides pre-heated ventilating air to the building, which ultimately escapes through cracks and openings. If warm air leaks out of the collector array, cold air must enter the building, thus increasing the normal amount of infiltration and discomfort.

Ducts may be made of fiberglass ductboard or sheet metal with insulation on the inside or outside, with external insulation usually being preferred. Ductboard can be taped to seal the joints, and metal ducts may be joined with drive clips and hard casted or covered with duct tape. Bends and elbows should contain turning vanes to minimize pressure losses and fan power consumption.

Connections between ductwork and blowers should be made with flexible (usually fabric) sections to dampen noise from vibrating blowers and motors, and for ease of removing the blower for service and maintenance. Connections between the ducts and storage box should be flexible to allow for differential movement of the storage box relative to the ducts.

ADAPTING TO EXISTING HEATERS

Solar heating systems are readily adaptable to air distribution systems. While baseboard convectors are prevalent in many non-solar hydronic systems, flat-plate collectors will not perform well with such systems so fan-coil units are usually preferred. A water-to-air coil can be installed in the cold air return duct of an existing warm air-heating system as illustrated in Figure 14-3. Usually the blower is an integral part of a furnace, whether fuel-fired or electric, and for electric furnaces the automatic controls must be reconnected so that the

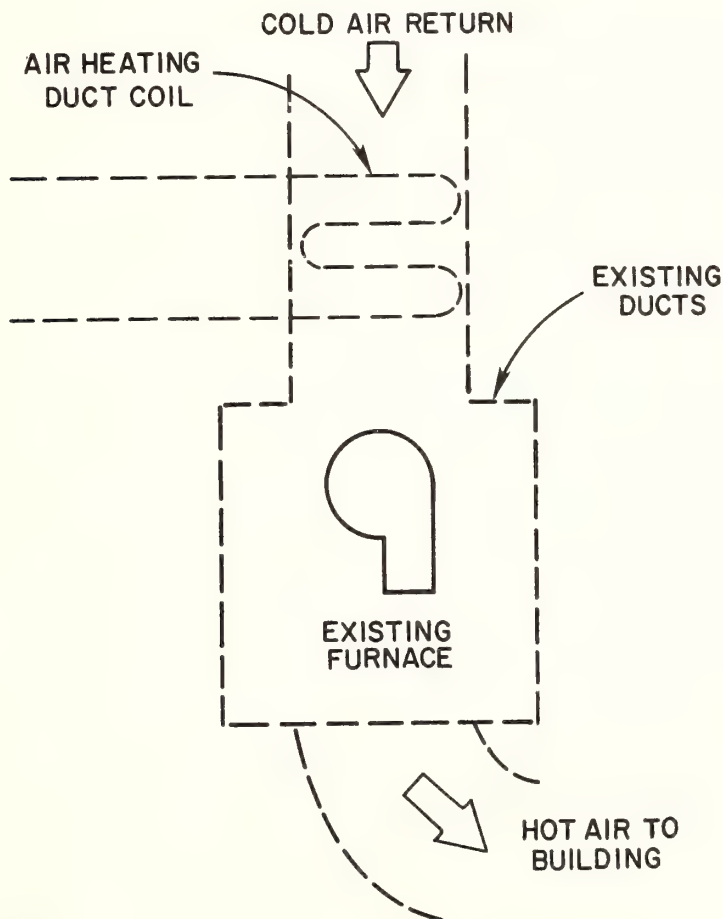


Figure 14-3. Recommended Arrangement for Retrofit Installations with a Central Air Circulation System

blower is independent of the heating element. Locating the heating coil ahead of the blower, as shown in Figure 14-3, is preferred so that air is pre-heated by the solar system. However, if the blower motor is in the air stream, its life may be reduced because it is subjected to a high temperature environment. Alternatively a type B (higher temperature operation) motor may be used. Generally, the location of the water-to-air heating coil is dependent upon the duct arrangement.

A two-blower arrangement for a retrofit air solar system might be arranged as shown in Figure 14-4. The existing blower will have to be decoupled from the heating element control and reconnected to the central solar system controller.

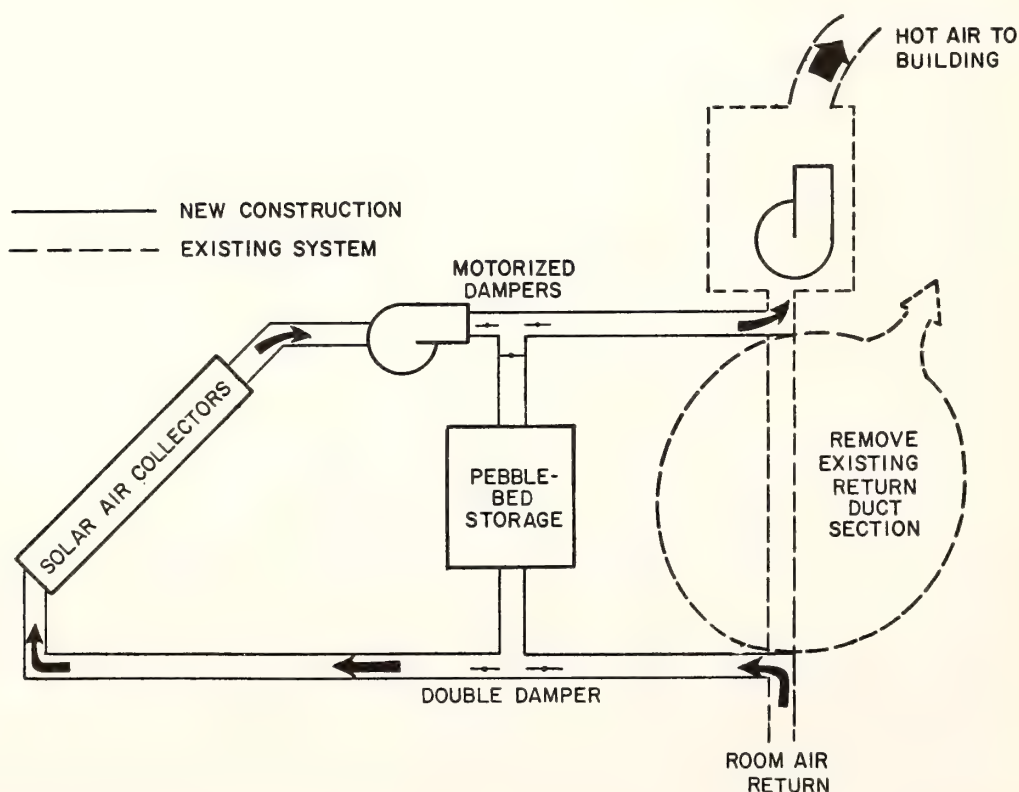


Figure 14-4. Air-Heating Solar System for Retrofit Installations

ELECTRIC SERVICE

Connections to the control box, pumps, and blowers are usually 120 V A.C. single-phase power. Electrical connections to valves and damper motors are normally 24 volts, and transformers are provided with the controller. Low voltage wiring is also appropriate for thermostats.

Control panels should be located conveniently for easy access during installation and maintenance. Control devices are usually field-wired according to instructions provided by the system (or controller) manufacturer.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 15

OPERATIONAL CHECK-OUT

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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OBJECTIVE

The objective of this module is to acquaint the trainee with:

1. The importance of thorough check-out of a completed solar heating system.
2. The general procedure for checking the proper functioning of the system.
3. The information for adjusting the controls in a solar heating system to achieve effective and dependable operation.

INTRODUCTION

The numerous components in a solar heating system, with their interconnection and interdependence, require thorough checking for proper functioning before final completion and acceptance. Although there are substantial differences in systems, a reasonably standard procedure of inspection, functional checking, and adjustment can be applied in most installations. Where check-out instructions of the system designer and/or manufacturer are available, they should be closely followed. Step-by-step directions are included in installation manuals provided by a few suppliers of complete solar heating systems. Specific directions for installation, adjustment, and checking are also usually supplied by manufacturers of solar system controllers. The installer's use of a complete check list or form is recommended as the best protection against omissions, subsequent operation problems, customer complaints, expensive damage, and costly service calls.

GENERAL PROCEDURE

VISUAL INSPECTION

The first step in a practical check-out procedure is a visual comparison of the completed system with construction drawings and manufacturers' instructions. Fluid flow patterns, sizes and locations of ducts and piping, and sealing of duct joints should be verified. Electrical connections on controls, sensors, and motors should be checked for integrity and conformity with plans. The collector installation should be inspected for alignment, tight connections, properly positioned piping and ductwork, and security of collector support and perimeter flashing.

After completion of the visual inspection of the idle system, various observations should be made when the equipment is operated in each required mode. Generally, the following checks will usually be necessary:

1. Observation of leakage of liquids or air at any point in the system, while running in each operating mode;
2. Verification of proper functioning of all moving parts and controls, including motors, pumps, fans, motorized valves and dampers, manual valves and dampers, check valves and dampers, fuel valves, electrical switches and relays, thermostats and their contacts, draining and venting valves and fittings, and any other adjustable components in the system;
3. Liquid levels and compositions;

4. Cleanliness and freedom from obstructions in flow channels for liquids and air;
5. After initial operation, cleanliness of filters in liquid and air circuits;
6. Verification of proper control settings and their proper action (turning on, turning off) in actuating motors, valves, and dampers;
7. Detailed check of proper functioning of safety and limiting devices and designs, such as complete drain-down of water from collectors and piping in drain-down systems, automatic valves for draining, venting, and bleeding liquid systems, overheat protection devices such as pressure relief valves and sensor actuated drain valves, and any other system protection devices;
8. Complete check of auxiliary heating and distribution system.

The check-out procedure outlined above should be followed prior to final insulation of the principal system components and piping unless insulation is part of the equipment such as in internally insulated air ducts. The heat storage unit may, however, be insulated prior to final system check.

FINAL CHECK

Following the testing and the repair and correction of faults which may be found, insulation should be applied to components and interconnections in the system, as recommended or required. A final check on the proper functioning of all moving parts and control functions should then be made to eliminate possibility of malfunction accidentally caused

by damage during the insulation process, and as a double check on system functions. This final check-out may advantageously be done in collaboration with the system owner so that he may understand its operation.

OPERATIONAL CHECK-OUT

Of particular importance is the careful use of manufacturer's and designer's check-out instructions in verifying and correcting the installation and operation of each system and its components. Procedures are provided by some suppliers of complete solar heating systems, and the manufacturers of the principal components usually supply check-out information on their own products. Collectors, pumps, motorized valves and dampers, and control subsystems are examples.

In checking operations in various modes, it is usually necessary for the system installer to simulate one or more conditions not prevailing at the time of testing. If the sun is obscured, for example, solar collector checking requires imposing some type of artificial condition to actuate the appropriate components. The supplier of the controller or of the complete system usually offers instructions on simulating each operating mode by making jumper connections across sensor terminals. Sensor control settings must be verified, however, by observations under actual conditions.

PERFORMANCE TEST

The third major checking procedure is the measurement of heat delivery from the solar heating system. Although only a few systems are provided with convenient and accurate facilities for solar system efficiency checking, the installer can usually make a reasonably reliable

determination of performance. By comparing the measured results with design values, the quality of the system and its installation can be verified. As indicated in more detail below, measurement of inlet and outlet collector temperatures, and measurement or estimate (based on measured pressures) of collector fluid flow rate near noon on a sunny day will provide sufficient information for at least a minimum evaluation of system performance.

A general blank form for use in checking a complete solar heating system is presented in the Appendix to this module. It includes numerous items which might be considered trivial or obvious, but even a minor fault, if overlooked and uncorrected, can cause poor performance, system deterioration, or building damage. Heating practitioners may also notice omissions which, in their experience, should be covered in the check-out process. Since the listing is intended to apply to all types of solar heating systems, some items are not applicable in each case considered.

Although sometimes difficult, measurement of solar heat collection and delivery to use provides the best assurance of good system performance. Fluid temperature rise and flow rate through the collectors, ambient temperature, and solar radiation should be measured. Solar heat collection and collection efficiency can then be calculated. If flow rates cannot be directly measured, they may sometimes be approximated by determining pressure drops across manufactured standard equipment, such as a heat exchanger or collector, in which the pressure-flow relationships are known.

AIR SYSTEMS

Among the check-out steps listed in the general procedure, several are particularly important in air systems. Air leakage, damper closure, blower and motor operation, and proper control should be thoroughly checked. Because of their specialized aspects, collectors, storage units, air handlers, and controllers require more than routine attention.

The collector inspection check list used by installers of a nationally distributed solar air heating system follows:

INSPECTION CHECK LIST - COLLECTORS⁽¹⁾

1. Collector Array: Refer to plans
2. Holding proper dimension from collector to collector - thus making airtight seal from port to port: Refer to specifications and/or plans.
3. Used specified material for port and end cap sealant: Refer to specifications and/or plans.
4. Relief tubes sealed and in place: Refer to specifications.
5. Confirm location and dimensions of 2 x 8 (1½" x 7 1/8") frame.
6. Cap strips installed so proper and airtight seal is accomplished.
7. Perimeter insulation installed: Refer to specifications and/or plans.
8. Perimeter flashings installed properly: Refer to specifications and/or plans.

⁽¹⁾Courtesy of The Solaron Corporation, used by permission.

9. Connecting Collars: Refer to plans.
 - a. Location
 - b. Sealed properly
10. Heat Sensors: Refer to plans.
 - a. Installed properly
 - b. Correct location

An inspection list for the pebble-bed heat storage unit specified by the same solar air system manufacturer follows:

INSPECTION CHECK LIST - PEBBLE-BED HEAT STORAGE UNIT⁽¹⁾

1. Location: Refer to plans.
2. Location of Duct Openings: Refer to plans.
3. Dimensions of Unit: Refer to plans.
4. Dimensions of Duct Openings: Refer to plans.
5. General Construction: Refer to plans.
 - a. If construction does not follow plans, make sure modifications are adequate to meet specifications
 - b. Check for exposed wood - all combustible surfaces must be covered with a non-combustible material
6. Check all joints:
 - a. Sealed adequately
 - b. Correct sealant used (i.e. suitable for temperatures around 180°F)

⁽¹⁾Courtesy of The Solaron Corporation, used by permission.

7. Lower Plenum: Refer to plans.
 - a. Proper materials used
 - b. Correct dimensions
 - c. Correct spacing of bond beam block
8. Upper Plenum: Refer to plans.
Correct Dimensions
9. Rock: Refer to specifications.
 - a. Proper size
 - b. Free of foreign materials (clean)
 - c. Proper amount
 - d. No depressions in rock bed (level on top)
10. Storage Unit Lid: Refer to specifications.
 - a. Construction
 - b. Sealed adequately
 - c. Proper sealant used (i.e. suitable for temperatures around 180°F)

An air handler start-up procedure, which is effectively an installation check list, is shown below. This procedure also verifies the proper functioning of nearly all the elements in the control system.

START-UP PROCEDURES - AIR HANDLER⁽¹⁾

I. Preliminary Check

1. Double check all line and low voltage wiring and connections (see wiring diagram for exact wiring hook-up to unit).

⁽¹⁾Courtesy of The Solaron Corporation, used by permission.

2. Check all damper positions, both inside air handler and any dampers that might be located elsewhere in the duct system.
3. Check belt-driven power train (tighten set screws on pulleys, confirm V-belt alignment, etc.).
4. Check voltage supply to unit. Should voltage not be correct, contact electrician before proceeding with start-up.
5. Open all registers, diffusers and grilles in distribution system.

II. Start-up on "Sunny Days"

1. Set heat anticipators in space thermostat.
 - a. W_1 (first stage heating) set at 0.7 amp.
 - b. W_2 (second stage heating) set at 0.1 amp.
2. Set thermostat so W_1 is calling for heat. W_2 must not be calling for heat at this time.

NOTE: The use of a jumper between W_1 and R_H at the AU control panel can be used in lieu of setting the thermostat. Before the system is given approval, however, check system operation with the thermostat to insure proper operation.

3. Set Sub-base switches (if present).
 - a. "Fan-Auto"
 - b. "System-Heat"
4. Turn on circuit breakers.
5. Turn on disconnect feeding the AU air handler.

6. Observe unit operation.
 - a. The auxiliary furnace blower should be running. There should be no auxiliary heat.
 - b. The AU air handler blower should start as long as sensors T_{co} and T_{ci} have a 45 degree F or greater temperature differential.
 - c. Air flow through dampers in the air handler, and BD1 and BD2, should be as shown on your plans (refer to A.E.M. "Control System" schematic).
 - d. A temperature differential of less than 45 degrees F will automatically switch the system into a "heat from storage" mode. If there is no heat in storage (less than 90 degrees F), the control board will automatically by-pass the solar heating circuit and bring on the auxiliary heat source without having W_2 in the space thermostat in a "heat" position (closed circuit).
 - e. When the rock storage unit has enough heat (greater than 90 degrees F) available, and the T_{co} , T_{ci} differential is less than 45 degrees F, the air handler will direct the air flow through the rock storage unit and into the auxiliary furnace. This will be accomplished without the auxiliary heat coming on.
7. Set the space thermostat so W_1 and W_2 are not calling for heat (open).
 - a. The auxiliary furnace blower will cease operating
 - b. The AU air handler will continue to operate

- c. The dampers inside the AU air handler will direct the solar heated air to the rock storage unit (see flow schematic on your plan and/or your A.E.M. "Control System" schematic.)
- 8. Set the space thermostat so W_1 and W_2 are calling for heat (closed circuit).
 - a. The auxiliary furnace should operate in a conventional manner
 - b. The AU air handler blower will continue to operate
 - c. Dampers in the AU unit will direct air through the rock storage and into the auxiliary furnace (see flow schematic on your plan and/or "Control System" schematic in your A.E.M.).

PERFORMANCE CHECK

Knowledge of the heat output of a solar air heating system is of value to the owner. If measurement of air flow rate is impossible or impractical, static pressure readings at several places in the system can be used to obtain a reasonably close estimate of flow rate by comparing measured pressure drop across collectors, air handler, or hot water coil against manufacturer's data. Small holes (1/4-inch diameter) may be drilled or punched through duct walls at the positions shown in the Solar Heating Flow Schematic shown on the next page⁽¹⁾. Flexible tubing is then inserted for connection to a manometer.

⁽¹⁾Courtesy of The Solaron Corporation, used by permission.

PROJECT NAME: Typical Project DATE: _____

PROJECT TYPE: RES'L X COMM'L _____ IND'L _____

AGRI _____ OTHER _____

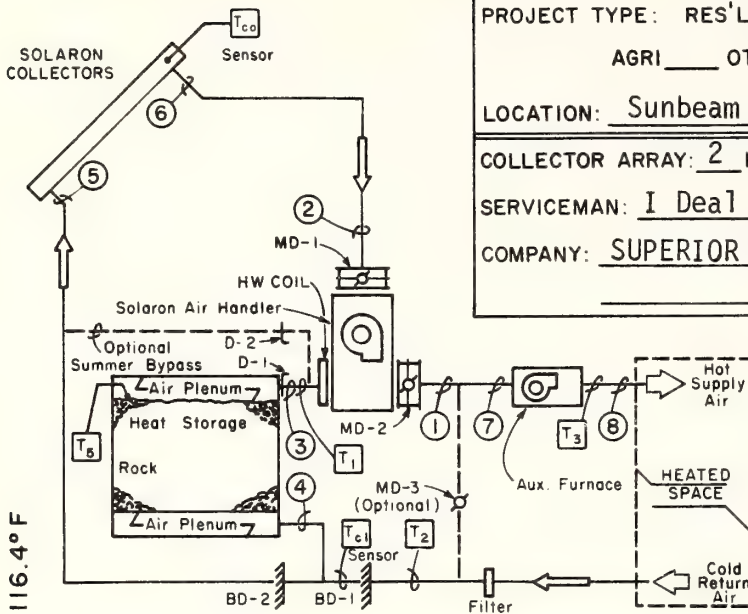
LOCATION: Sunbeam Valley, California

COLLECTOR ARRAY: 2 HIGH X 12 WIDE = 468 sq.ft

SERVICEMAN: I Deal Heavenly

COMPANY: SUPERIOR SUNBEAMS, INC.

PHONE () _____



SOLAR HEATING FLOW SCHEMATIC

SEQUENCE OF OPERATIONS										
	MD-1	MD-2	BD-1	BD-2	MD-3	D-4	Pump	Solaron air handler	Aux. Furnace	
						D-2		Fan	Heater	
HEATING FROM COLLECTOR	O	O	O	O	P.O.	O	ON	ON	ON	OFF
HEATING FROM STORAGE	C	O	O	C	P.O.	O	OFF	OFF	ON	OFF
STORING HEAT	O	C	C	O	O	O	ON	ON	OFF	OFF
HEATING WITH AUX. FURNACE	C	O	O	C	P.O.	O	OFF	OFF	ON	ON
WATER HEATING (SUMMER)	O	C	C	O	O	O	ON	ON	OFF	OFF

FOR HEAT PUMP SYSTEMS: MD-2 Closed, MD-3 Open, BD-1 Closed

$$T_3 = \frac{T_1 (CFM_{sol}) + T_2 (CFM_{aux.} - CFM_{sol})}{CFM_{aux.}}$$

$$T_1 = 130^\circ\text{F}$$

$$T_2 = 68^\circ\text{F}$$

$$T_3 = 117^\circ\text{F}$$

	AIR HANDLER	
	SOLAR	AUX
Design CFM	936	
Design Ext. SP	0.80"	
Fan RPM	1160	
HP	1/2	
Motor RPM	1725	
Volt	115	
Phase	1	
FLA	6.0	
SF	1.25	
SFA	6.8	
Insul. Class	B	
Motor Mfg	GE	
Model No	MT0051	

TEMPERATURE & STATIC PRESSURE MEASUREMENTS

STORING HEAT				HEATING FROM STORAGE				HEATING FROM COLLECTOR			
Motor Amps. 5.0								Motor Amps. 5.3			
POINT	°F	STATIC PRESSURE	S.P. DIFF.	POINT	°F	STATIC PRESSURE	S.P. DIFF.	POINT	°F	STATIC PRESSURE	S.P. DIFF.
1		0.00"		1	-	0.00"		1	-	-.19"	.42"
2	138	-.34"	.69	2	-	-.01"		2	-	-.94"	
3	132	+.35"	.19	3	130	-.29"	.14	3	-	-.20"	.04"
4	68	+.16"		4	68	-.15"		4	-	-.16"	
5	66	+.06"	.31	5	-	-.01"	.00	5	66	-.34"	.42"
6	141	-.25"		6	-	-.01"		6	122	-.76"	
7				7	-	-.42"	.60	7	-	-.21"	.48"
8				8	117	+.18"		8	-	+.27"	

PERFORMANCE ANALYSISAir Heating System

A. Collector

1. Area, A_c _____ ft^2
2. Inlet temperature, T_i _____ $^{\circ}\text{F}$
3. Outlet temperature, T_o _____ $^{\circ}\text{F}$
4. Pressure drop across blower, W_p _____ in W.G.
5. Flow rate, \dot{V} _____ cfm
6. Air density, ρ _____ $.07 \text{ lb/ft}^3$
7. Specific heat, c_p _____ $.24 \text{ Btu/(lb}\cdot^{\circ}\text{F)}$
8. Heat delivery rate:

$$Q_c = (\dot{m}c_p)(T_o - T_i)(60)$$

$$= \dot{V} \cdot \rho c_p (T_o - T_i)(60) \quad \text{_____ Btu/hr}$$

B. Collector Efficiency:

1. Solar radiation on horizontal surface, I_H _____ $\text{Btu}/(\text{ft}^2 \cdot \text{hr})$
or on tilted surface, I_T _____ $\text{Btu}/(\text{ft}^2 \cdot \text{hr})$
2. Collector efficiency

$$h_c = \frac{Q_c}{I_T A_c} \times 100$$

$$\text{_____ \%}$$

or sensitive differential pressure gauge. Temperature sensors (glass or dial thermometers, thermocouples, or thermistors) are also inserted through holes at these points.

A performance analysis form used by installers of a widely used air system, with sample data, is shown on the preceding page⁽¹⁾. Points for measurement of temperature and static pressure are indicated. By comparison of static pressure differences across the several components with those shown in the manufacturer's engineering data sheets at various flow rates, actual flow rates may be closely estimated. Heat output may then be calculated by completing the form entitled Performance Analysis, Air Heating System. By use of a simple hand-held meter, solar radiation can also be measured so that collection efficiency, item B2 in the Performance Analysis form, may be calculated. Comparison with manufacturer's performance data may then be made. If a large difference is found, causes should then be investigated.

LIQUID SYSTEMS

The principal components of a typical liquid space heating system requiring check-out procedures specific to the liquid type (as compared with air) are

1. liquid pumps
2. liquid-to-liquid heat exchanger
3. valves and piping for draining and venting.

⁽¹⁾ Courtesy of The Solaron Corporation, used by permission.

The other system components such as collectors, storage tank, motorized valves, pipe connections, sensors and controllers, all have their counterparts in the air systems previously described. If the supplier of a main component, such as the collector, has provided a complete system design and check-out procedure, those instructions should be followed. In other instances, particularly if large or custom-designed systems are involved, the mechanical engineering designer or consultant may specify a check-out procedure. If no specific inspection list is available, the installer should follow the general guidelines previously outlined, altering them as may be necessary for the specific system involved.

PUMPS

With respect to pumps, the installer should check speed (unless directly coupled to motor), and by means of permanent or temporary gauges, the pressure difference from inlet to outlet of the pump. Proper mounting, alignment, and attachment to piping and wiring should also be checked. Noise and vibration should be within acceptable limits.

By use of the pump speed and the fluid pressure difference, the pump manufacturer's charts or graphs may be used to determine volumetric flow rate. This rate should then be compared with the desired or designed flow rate and, if not in satisfactory agreement, causes for difference should be determined and corrected.

Because misleading pressure readings may sometimes be obtained at points near pumps, measurements of pressure differences across other components in the system should also be made. Pressure loss through the collector array and across a heat exchanger may be compared with the

manufacturer's flow rate-pressure drop data to confirm the flow measurements. If serious discrepancies appear, their cause should be determined.

HEAT EXCHANGER

In dual-liquid systems, proper operation of the collector-storage heat exchanger must be verified. Counter flow of the two liquids is essential, so piping connections must be checked to verify the flow of one liquid in a direction opposite to the flow of the other. When operating under conditions such that solar energy is being collected, the inlet and outlet temperature of the collector fluid, the inlet temperature of the storage fluid, and the static pressure at all four points should be measured. If thermometer wells are not provided in the piping, temperature sensors (thermocouples, thermistors or small-bulb thermometers) should be tightly taped to the outside of the pipe in close proximity to the heat exchanger connections. At least 2 inches of pipe insulation should be applied over the temperature sensor along at least 1 foot of pipe length. After they become constant (several minutes) the readings will be sufficiently close to the liquid temperatures at those points. Calculation of a heat balance on the exchanger and a comparison of pressure loss with those shown in the manufacturer's data should then be made.

ANTIFREEZE SOLUTION

The concentration of antifreeze solution in the collector liquid should be verified by use of a hydrometer such as commonly available

for testing automobile radiator coolants. If the test shows inadequate freeze protection, addition of ethylene glycol or propylene glycol as specified by the designer should be made until the concentration is satisfactory. Tables of antifreeze concentration and liquid density may be used if the hydrometer does not show the freezing temperatures directly. Drainage of some liquid from the collector loop may be necessary in order that antifreeze can be added.

LIQUID LEVELS

The liquid level both in the collector loop and in the storage tank should be determined by whatever means are provided in the system. Sight gauges, pointer type indicators, inspection ports, and overflow valves or other means may be used, depending on design.

In a closed collector loop, proper filling and functioning of the expansion tank must be verified, sufficient liquid being present to fill the collector loop and sufficient additional space being available for the enlarged liquid volume when solar heated. Consideration must be given to the temperature of the liquid at which the expansion tank level is noted so that either an increase or decrease in liquid volume can be accommodated.

If a non-aqueous liquid is used in the collector loop, such as a silicone oil, its quantity and quality should be checked. Normal procedure would involve filling of the system by the installer, from a known supply of the chemical. In case of doubt, some chemical or physical property of the liquid which the manufacturer recommends for identification should be ascertained.

CORROSION INHIBITOR

If a corrosion inhibitor is used in the storage liquid, the most satisfactory procedure for insuring protection is careful compliance with designer's instructions when the inhibitor is first added to the system. If later verification of the quantity and quality of the additive is required, a chemical or physical test of a sample of the solution should be made by the installer. In some cases, a sample may have to be supplied to a testing laboratory for checking. In exceptional cases, the most economical procedure may be the draining and recharging of the storage unit, with properly measured additives.

DRAIN-DOWN SYSTEMS

Collector Loop

In drain-down systems, careful inspection of all piping and connections must be made to verify the absence of any low points or traps in the system that might prevent complete drainage. Even, horizontal runs of pipe should be avoided, slight slope being a much preferred design. Access of air to the collector either through an atmospheric valve or from the storage tank via an adequately sized pipe must be verified.

Systems that are designed for collector drainage to occur either when freezing threatens or when the pump ceases operation must be thoroughly and completely checked by dependable and repeated drainage and refilling, without fault. Those that are designed to drain back into the storage tank, either through an open return or siphon return

when the pump ceases operation, can be checked by interrupting the power supply to the pump motor. After a minute or so, absence of water outflow through an opened drain valve located in the collector supply piping above the level of water in the storage tank indicates satisfactory collector drainage. The lack of water in the return line from the collector to the storage tank can be similarly verified. In siphon return systems, the opening of the "siphon breaker" air inlet valve at the top of the collector piping assembly must also be verified. This valve must be open whenever electric power is accidentally or purposely interrupted.

Start-up of the system after drainage must also be checked by restoring power to the pump and to the siphon breaker valve (if the valve is of the electric-operated type). Observation of flow returning to storage, if in a visible location, or verification of normal pressures at various points in the circulating loop can confirm satisfactory displacement of air from the collector and piping, and the restoration of normal flow. In the siphon return system, verification of proper functioning of the air bleed valve at the top of the piping array must also be made.

Storage Tank

If main storage is operated at a pressure greater than atmospheric pressure and is non-vented (usually because a pressurized hydronic heat distribution system is involved), proper operation of the pressure relief valve (safety valve) on the storage tank must be established. If the valve cannot be tested satisfactorily in place, it should be removed

temporarily and tested with measured pressure by use of a pressure test kit. The greater complexity of the piping and valving in this type of system requires verification of complete collector drainage by methods recommended by the designer of the system employed. There is sufficient variation in the designs of these systems, including liquid level control, automatic water make up, check valves, by-pass piping, and motorized valves, that there is no standard check-out procedure applicable to all systems.

Sensors and Controls

Systems which involve collector drainage only when freezing threatens are usually actuated by a temperature sensor in the collector or in the atmosphere. These controls must be checked by artificially cooling the sensor below the temperature at which it functions to drain the collectors. The application of ice to the sensor should turn off the pump and allow the collector to drain. Removal of the ice will then permit the sensor to warm up and restart the pump. These operations can be visually verified.

Proper functioning of controls and other components which prevent overheating and/or boiling of fluids in the collector and storage loops must also be verified. Several types of overheat protection devices are commonly used, so the manufacturer's instructions should be carefully followed in the check-out procedure. An excessive temperature condition can usually be electrically simulated by a suitable signal to the controller, and the functioning of the overheat protection system observed. Circulation of water from the storage tank through a heat rejection

coil, drainage of the collector, draw-off of domestic hot water from the solar pre-heat tank and automatic addition of cold water, and discharge of hot water from the solar hot water storage tank, are the principal methods employed.

Checking the operation of whatever system is in use is essential. If the overheat sensor is accessible, a small electric heating element can be temporarily applied at the proper point by the installer, thereby checking the entire system in place, including the control elements. If this condition has to be simulated by causing an open circuit or short circuit in the overheat sensor contact in the controller, the sensor itself should be checked by heating prior to installation. Controller manufacturers usually provide check data for all standard elements in the control circuitry. Measurement of electrical resistances can usually provide satisfactory evidence of proper operation.

PERFORMANCE CHECK

Following the checking of proper functioning of all components in the system under all conditions of operation, measurement of heat output and efficiency of the system should be made. The following table (Performance Analysis, Liquid Heating System) indicates the measurements needed and the calculations required. The previously described methods for measuring collector flow rate and temperature rise at full and nearly constant solar radiation levels should be conducted simultaneously with measurement of solar radiation intensity. These measurements should be made near mid-day so that conditions are as constant as practical. Calculation of the heat delivery from the collector is a

simple multiplication of the temperature rise, flow rate, and heat capacity factors. Dividing this quantity by the solar radiation input rate provides efficiency data. Comparison with the manufacturer's rating data at the temperatures and solar radiation levels corresponding to those applied during the test can then show the degree of agreement with the designed values. Unexplained differences, if any, then need to be investigated and corrected.

PERFORMANCE ANALYSISLiquid Heating System

Collector:

1. Area, A_c _____ ft^2
2. Inlet temperature, T_i _____ $^{\circ}\text{F}$
3. Outlet temperature, T_o _____ $^{\circ}\text{F}$
4. Pressure drop across pump _____ psi
5. Fluid flow rate, G _____ gpm
6. Specific heat, c_p _____ $\text{Btu}/(\text{lb} \cdot ^{\circ}\text{F})$
7. Fluid specific weight, γ _____ lb/gal
8. Heat delivery rate:
 $Q_c = G\gamma c_p (T_o - T_i)(60)$ _____ $\frac{\text{Btu}}{\text{hr}}$

Collector Efficiency:

1. Solar radiation on horizontal surface, I_H , or
 tilted surface, I_T _____ $\text{Btu}/(\text{ft}^2 \cdot \text{hr})$
 _____ $\text{Btu}/(\text{ft}^2 \cdot \text{hr})$
2. Collector efficiency
 $\eta_c = \frac{Q_c}{I_T A_c} \times 100$ _____ %

APPENDIX

Inspection Check List

Project: _____

Owner: _____

Address: _____

Date Inspected: _____

Inspector: _____

Approved: _____

Disapproved: _____

Exceptions: _____

(refer to Exception Notes)

Yes No

A. Solar Collectors:

- | | | |
|---|-------|-------|
| 1. Structural strength appears adequate | _____ | _____ |
| 2. Are resistant to weather | _____ | _____ |
| 3. Are resistant to fire | _____ | _____ |
| 4. Provided with an efficiency curve in acceptable form | _____ | _____ |
| 5. Have material with potential for outgassing | _____ | _____ |
| 6. Are provided with adequate flashing | _____ | _____ |
| 7. Meet safety requirements | _____ | _____ |
| 8. The glazing is adequate for strength | _____ | _____ |
| 9. The glazing appears to have satisfactory durability | _____ | _____ |
| 10. The absorbed material and coating are acceptable | _____ | _____ |

B. Energy Transport System:

- | | | |
|---|-------|-------|
| 1. Expansion or contraction will cause damage | _____ | _____ |
| 2. Joints with dissimilar metals are suitably protected | _____ | _____ |
| 3. Air distribution system is adequately sized | _____ | _____ |
| 4. Filters are placed properly | _____ | _____ |
| 5. Pipe and duct hangers are adequate | _____ | _____ |
| 6. Valves and dampers are acceptable | _____ | _____ |

	<u>Yes</u>	<u>No</u>
C. Thermal Storage Units:		
1. Container materials are sufficiently durable	___	___
2. Contamination of air or water is adequately prevented	___	___
3. Materials in storage units are compatible with other units of the system	___	___
D. Heat Transfer Fluids:		
1. Are chemically stable	___	___
2. Are thermally stable	___	___
3. Are non-corrosive	___	___
4. Pose a safety hazard	___	___
5. Have a sufficiently high flash point	___	___
E. Heat Exchangers:		
1. Are suitable with the system	___	___
2. Potable water is suitably protected from non-potable heat source	___	___
3. Heating coils are adequate	___	___
F. Gaskets and Seals:		
1. The system is adequately gasketed for pressure	___	___
2. Ducts are adequately sealed	___	___
G. Insulation and Moisture Protection:		
1. Insulation materials constitute a fire hazard	___	___
2. Materials are thermally stable	___	___
3. Pipe and duct insulation is adequate	___	___
4. Storage is adequately insulated	___	___
H. Hose Couplings:		
1. Are thermally stable	___	___
2. Are compatible with the heat transfer fluid	___	___
3. Are compatible with piping material	___	___
I. Controls:		
1. Fail-safe protection is adequate	___	___
2. Pressure relief devices are adequate	___	___
3. Vacuum relief is provided	___	___
4. Main shut-off valves and switches are identified	___	___
5. Automatic controls are provided	___	___

	<u>Yes</u>	<u>No</u>
J. Installation, Operation and Maintenance:	—	—
2. The system is generally maintainable	—	—
K. Auxiliary Heating and Hot Water Units:		
1. The auxiliary heater is sized adequately to maintain comfort conditions	—	—
2. The auxiliary DHW system is adequate	—	—
L. Solar System:		
1. Leak tests have been performed	—	—
2. Freeze protection is adequate	—	—
3. Adequate isolation valves have been installed	—	—
4. There is adequate drainage when components are installed and dismantled	—	—
5. There is generally adequate protection from scalding temperatures	—	—
6. Catchments are provided for boil-out and overflow of toxic materials	—	—
7. Water hammer arresters are adequate	—	—
8. Air bleeds and vent valves are adequate	—	—
9. Vents, chimneys and ventilation are adequate	—	—
10. Performance estimates are available	—	—

Exception Notes

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 16

FUNDAMENTALS OF SOLAR COOLING

SOLAR ENERGY APPLICATIONS LABORATORY
COLORADO STATE UNIVERSITY
FORT COLLINS, COLORADO

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GLOSSARY OF TERMS

absorbent	A liquid in which a refrigerant can be dissolved or combined
coefficient of performance	Ratio of heat removal rate (cooling rate) to energy supply rate
refrigerant	Working fluid in a refrigeration system
ton of refrigeration	Heat removal at a rate of 12,000 Btu per hour

OBJECTIVE

The objective of this module is to develop understanding of the principles of solar space cooling systems. In order to test whether this objective is met by the trainee, as a minimum level of accomplishment, the trainee should be able to:

1. List the different solar cooling methods and
2. Describe the operation of solar cooling systems.

INTRODUCTION

The withdrawal of heat from the air within a building enclosure which results in a temperature or humidity lower than that of the natural surroundings is termed space cooling or air-conditioning. Cooling methods powered by solar energy are of particular interest in this module.

CATEGORIES OF SPACE COOLING METHODS

There are three categories of space cooling methods for residential buildings. They are:

1. Refrigeration
2. Evaporative cooling
3. Desiccant cooling (dehumidification).

Solar energy may be used as the principal energy supply in some refrigeration systems and in desiccant cooling cycles. Evaporative cooling is only indirectly related to solar in being dependent on

climatic factors and in the opportunity for joint use of some of the solar heating equipment. The discussion in this module concerns principally refrigeration methods. Evaporative and desiccant cooling are also briefly mentioned.

DEFINITION OF TERMS

The capacity of a refrigeration machine to cool room air is customarily measured in tons of refrigeration. A ton of refrigeration is the removal of heat at a rate of 12,000 Btu per hour. Another often-used term in connection with refrigeration equipment is coefficient of performance, COP. The COP expresses the effectiveness of a refrigeration cooling system as the useful refrigeration effect divided by the net energy supplied to the machine. COP is determined by the simple equation below:

$$\text{COP} = \frac{\text{Heat energy removed}}{\text{Energy supplied from external sources}}$$

The COP of a mechanical vapor-compression refrigeration machine is characteristically about two and can be as high as four. The COP of a lithium-bromide-water absorption refrigeration machine is about 0.8 and more often operates in the range from 0.6 to 0.7. A COP less than 1.0 means there is more energy supplied to the machine than heat energy removed from the room air. From the cooling capacity and COP, the energy consumed by the machine to produce the cooling effect can be determined by dividing the heat removal rate by the COP. For example, with a 3-ton absorption air chiller having a heat removal rate of 36,000 Btu per hour and a COP of 0.6, the rate of heat supply to the generator is 60,000 Btu per hour ($36,000 \div 0.6$).

REFRIGERATION SYSTEMS

Cooling is achieved in refrigeration systems by removing heat from air or water as it comes in contact with a cold refrigerated surface. That surface is maintained at its low temperature by evaporating a liquid refrigerant at a still lower temperature from the opposite side or face of the surface. Heat absorbed by the evaporating refrigerant is supplied by the air or water being cooled. The most common method for providing the energy necessary for this process is by an electrically powered compressor which raises the pressure of the refrigerant vapor so that it can be condensed to a liquid for subsequent evaporation at lower pressure and temperature as shown in Figure 16-1.

Conventional vapor-compression refrigeration systems powered by electric motors can be driven by electricity produced in a solar engine of some type, or the compressor can be powered directly by the solar engine, without electricity generation. A system of this type is described below.

Another process for supplying refrigerant to a surface through which heat is transferred from air or water being cooled involves the heating of a mixture of refrigerant and an absorbing liquid. The refrigerant is condensed and evaporated as in the vapor-compression system, then reabsorbed and returned to the generator where it is again vaporized by heat from fuel. These absorption refrigeration systems may be modified for solar heat supply. Further details are shown in Figure 16-2 and in the following sub-section.

Of the two principal types of commercially available refrigeration units, only the absorption type is driven by solar energy to an appreciable extent. Among several types of experimental and commercial

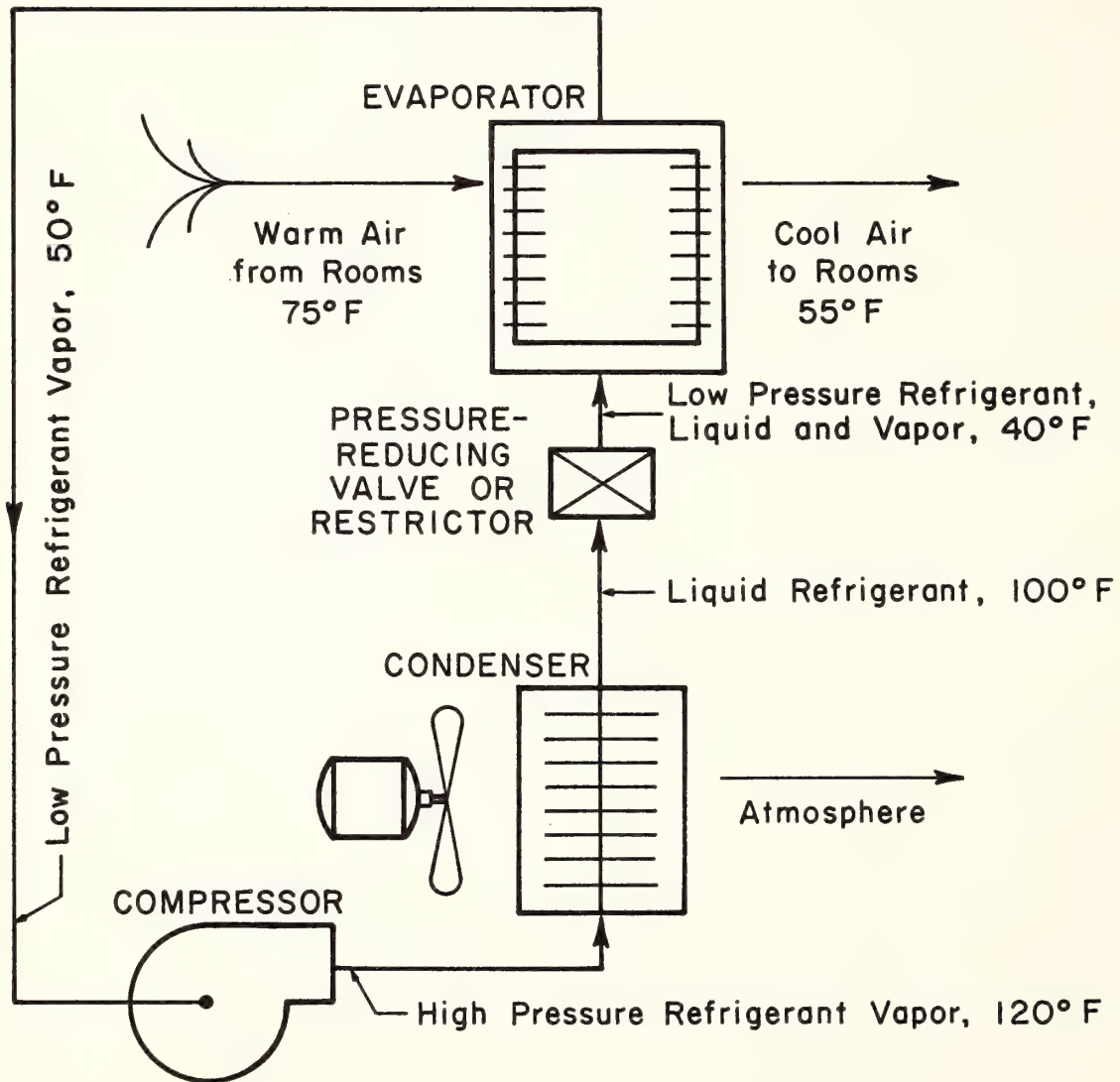


Figure 16-1. Vapor-Compression Air-Conditioner Schematic

which combines with or dissolves the refrigerant at low temperatures but will release the refrigerant when heated to higher temperatures. As the refrigerant is absorbed, heat is generated and must be removed by a cooling medium of some type, usually water.

Operation of a lithium-bromide absorption cycle is explained with the aid of Figure 16-2. Water vapor refrigerant is boiled from the liquid solution in the generator by providing solar heated water to the generator at temperatures between 160°F and 210°F. Vapor leaving the generator enters the condenser, where it is cooled to about 100°F and condensed to liquid by cooling water from an outdoor cooling tower. The condensate passes through a pressure-reducing valve or restrictor into the evaporator (chiller) coil where it cools to about 40°F and vaporizes by absorbing heat from the air or water being chilled. Vaporized refrigerant then passes to the absorber where it meets concentrated lithium-bromide solution from the generator at a temperature of about 100°F. In this absorption process, heat is released and removed by cooling water from the cooling tower. The lithium-bromide solution is returned from the absorber to the generator via a heat exchanger, by a pump or by gravity. The recuperator in the diagram is a heat exchanger which pre-heats the dilute solution as it flows from the absorber to the generator and at the same time cools the hot concentrated solution which flows from the generator to the absorber.

Temperature Restrictions

The temperature of the hot water supplied to the generator of a solar-operated lithium-bromide absorption refrigeration machine is usually between 160°F and 210°F. The heat input rate to the generator

must be sufficiently high to boil the refrigerant (water) from the solution. If the supply temperature is below 160°F, the heat transfer rate diminishes to a point too low for satisfactory operation. Low temperature in the recuperator can result in crystallization of lithium-bromide in the outlet tube leading from the recuperator to the absorber, eventually extending to the generator as water continues to be boiled off and the concentration of lithium-bromide increases. If the supply temperature is appreciably above 210°F, the heat transfer rate diminishes to a point too low for satisfactory operation (not usually possible because of temperature limitations in the solar collector and storage tank).

Cooling water temperature is dependent on atmospheric temperature and humidity and is generally a few degrees above the wet-bulb temperature. Design cooling water temperature is normally 80°F, but considerable variation is common.

Types of Lithium-Bromide-Absorption Refrigeration Systems

There are two types of lithium-bromide-water absorption refrigeration systems. In the direct expansion tube, air is circulated from the building to the evaporator coil, where cooling and dehumidification occur. In the chiller type, water is cooled by circulation through the coil. The chilled water is then pumped through a separate fan-coil unit through which room air is circulated. Chilled water may also be stored in one or two tanks coupled both to the chiller and the water-to-air fan-coil, thereby reducing fluctuations and intermittent chiller operation.

Figure 16-2 shows a mechanical pump for return of liquid from absorber to generator. This design is used in a commercial 3-ton chiller. A direct expansion 3-ton air-conditioner has employed a bubble-pump (similar to a coffee percolator tube) to lift the boiling solution from the generator to a vapor-liquid separator, so that the height change provides the pressure difference between generator-condenser on the high pressure side and evaporator-absorber on the low-pressure side.

SOLAR RANKINE-CYCLE ENGINE

Instead of driving the compressor of a vapor-compression refrigeration machine with an electric motor, an alternative source of power for the compressor is a solar-powered engine. Solar heat can be used to produce steam or other vapor to drive a turbine. The turbine may be coupled directly to the compressor of a refrigeration machine, as shown in a schematic drawing of a simplified system in Figure 16-3, or an electric generator and motor set may be used to couple the two systems.

Heat is supplied to a boiler from a solar collector. Fluid in the boiler is vaporized and the vapor drives the rotor of the turbine. The rotating shaft then drives a compressor in a vapor-compression refrigeration machine which produces the desired cooling effect. Vapor from the turbine is condensed and returned to the boiler. The regenerator is a heat exchanger to recover some of the heat from the vapor leaving the turbine. Working fluids include water (steam), several types of fluorocarbons, chlorinated hydrocarbons, and others. This machine is still in

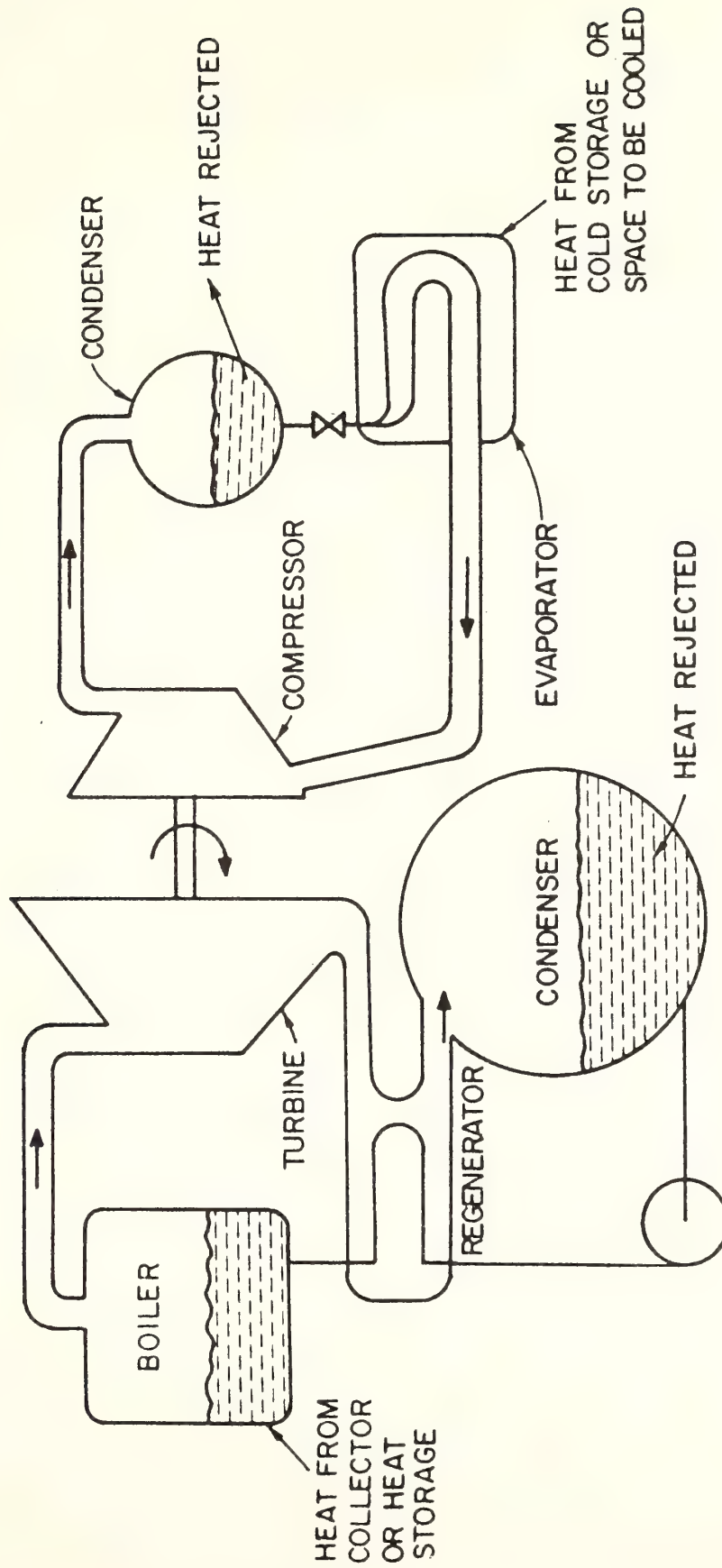


Figure 16-3. Rankine-Cycle Vapor-Compression System

experimental and developmental stages and not yet available as a commercial unit. If it becomes commercial, it is probable that this system would be more practical in large sizes (at least 25 tons) than in residential capacities.

HEAT PUMP

A heat pump can be used both for heating and for cooling. As a cooling unit, it is essentially a vapor-compression refrigerator which absorbs heat from inside a building and rejects it to the outside air. It is driven by an electric motor, normally supplied with commercial electricity from central station power plants. The principles of operation are described in Module 7.

EVAPORATIVE COOLING

EVAPORATIVE COOLING THROUGH ROCK BED

Air may be cooled by evaporation of water under conditions such that heat for vaporization is supplied by the air. The so-called swamp cooler employs this principle. As an example, ambient air at 100°F dry-bulb temperature and 70°F wet-bulb temperature (relative humidity 22 percent) can be evaporatively cooled by water sprays to about 77°F. However, the relative humidity would rise to an uncomfortable 71 percent.

It is evident that evaporative cooling does not require solar energy, so it is not a solar cooling process. However, because the rock bed in an air-heating solar system can be used for heat transfer and cool storage in an evaporative cooling system, this method is sometimes associated with solar cooling.

An evaporative cooler coupled with a rock-bed storage unit is shown in Figure 16-4. Night air is evaporatively cooled and circulated through the rock bed to cool the pebbles. During the day, warm air from the building can then be cooled by circulation through the cool pebble bed. Humidity supplied to the night air is thus not introduced to the living space. Dampers are positioned to direct the circulation of air appropriately. When cool air can no longer be delivered from the rock bed, outdoor air may be evaporatively cooled directly and delivered to the rooms. Evaporative cooling is practical only in fairly dry climates where relative humidities and night temperatures are normally low.

EVAPORATIVE COOLING WITH ROOF PONDS AND SPRAYS

There are two solar houses, one in California and the other in Arizona, that utilize evaporative cooling and night-time radiation to reduce day-time temperature rise in those structures. Each building has a shallow water pond on a flat roof with sectionalized retracting insulating covers over the pond. By retracting the covers at night, the water cools by evaporation and radiation to the sky. The covers are closed during the day to prevent solar heating of the pond. Heat is absorbed in the water by conduction through the ceiling of the rooms immediately below.

During the mild winters in these locations, the shallow ponds are used for space heating. The insulating covers are retracted during sunny days to collect solar heat in the pond and closed at night to prevent excessive heat loss. The stored heat is conducted through the metal roof (ceiling) for radiation into the living space below. In

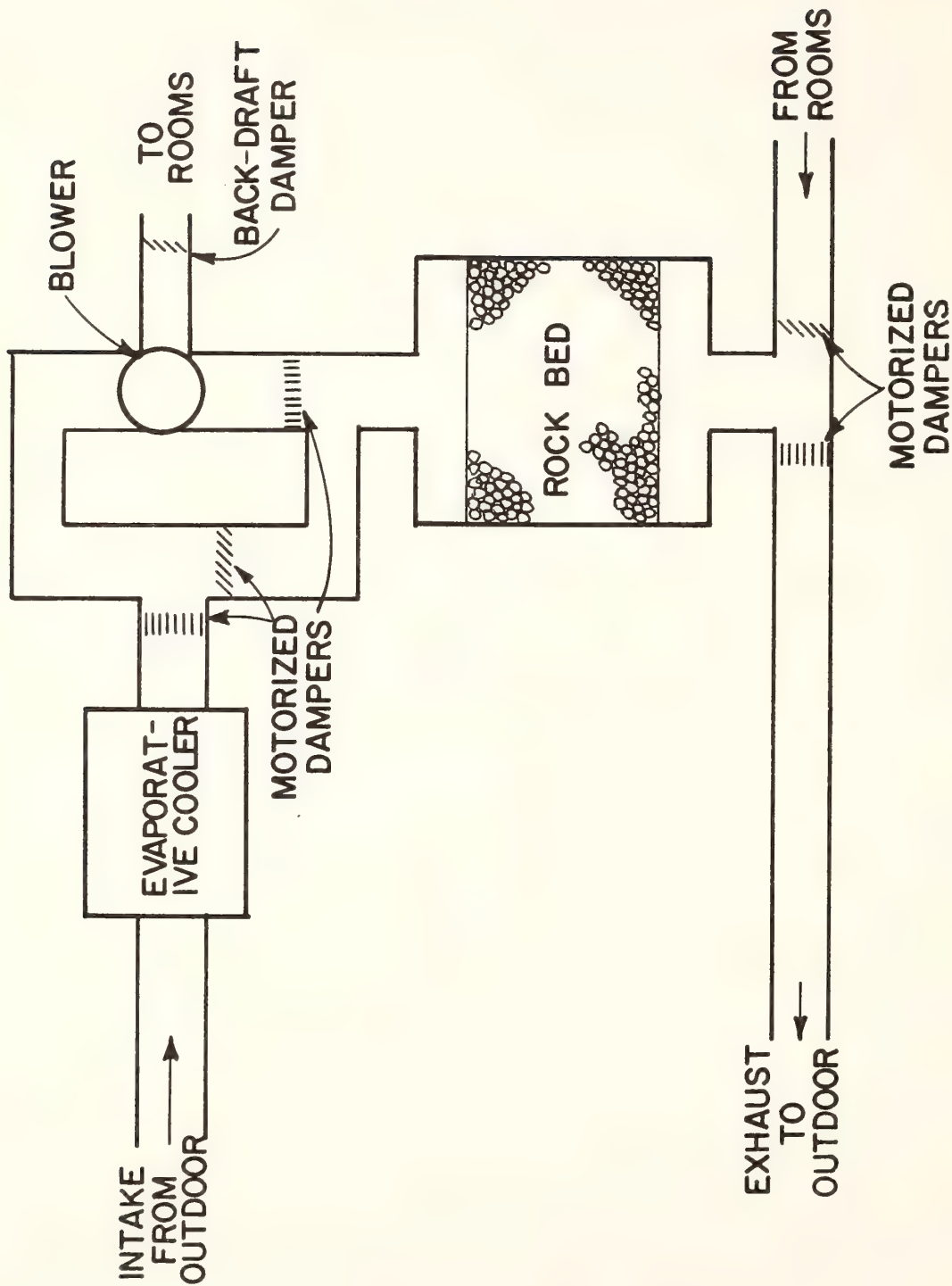


Figure 16-4. Evaporative Cooling with Rock-Bed Storage

climates where heating and cooling demands are low, this design may be of interest.

Roof ponds or sprays are also frequently used in commercial and industrial buildings simply to reduce excessive roof-top temperatures resulting from solar absorption on large area flat roofs. The exposed roof is continuously sprayed or flooded with water so that evaporation can minimize roof-surface temperature. Solar cooling is not involved.

DESICCANT COOLING (DEHUMIDIFICATION)

TRIETHYLENE GLYCOL OPEN-CYCLE DESICCANT SYSTEM

In locations where humidities are high, evaporative cooling can be accomplished if the air is first dehumidified by some means. One system commercially used for air dehumidification is shown schematically in Figure 16-5. Moist room air is dehumidified by contacting it with a solution of triethylene glycol in which moisture is absorbed. The dehumidified air then passes through mist eliminators and then through an evaporative cooler where its temperature is reduced to a comfortable level. The liquid desiccant passing through the absorber picks up moisture from the building air and becomes diluted. Solar heat may be used to regenerate the dilute triethylene glycol solution which is returned to the absorber and recycled. Regeneration is accomplished by spraying the glycol solution into a stream of solar heated air in which moisture evaporated from the glycol solution is exhausted to the atmosphere. If there is insufficient solar heat, an auxiliary heater may be used to heat the air stream. Heat is recovered from the glycol

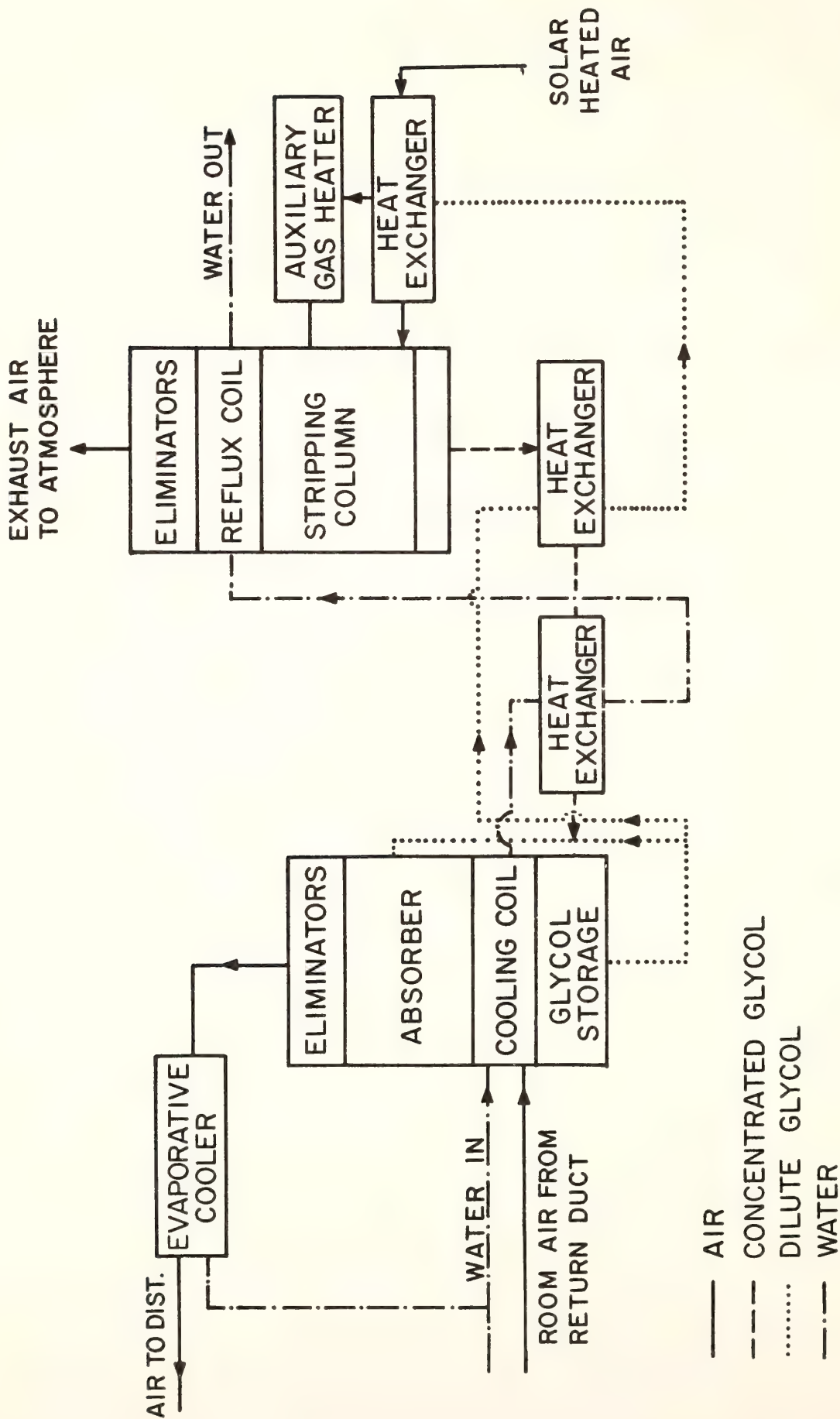


Figure 16-5. Schematic of Triethylene Glycol (Liquid Desiccant) Open-Cycle Air-Conditioning System

solution leaving the stripping column by heat exchange with solution leaving the absorber.

The system may be operated over a substantial range of air temperatures, but the higher temperatures, in the area of 180°F, result in superior system performance and higher COP's.

A liquid desiccant open-cycle system in large sizes, using conventional heat sources, is commercially available, but solar operation has been only in limited experiments. This type of system has not been actively considered for residential space cooling systems.

SOLID DESICCANT SYSTEMS

Another method for dehumidifying air in preparation for evaporative cooling involves use of solid adsorbents which can be regenerated by heat. Silica gel granules, lithium chloride and calcium chloride crystals, and "molecular sieve" zeolite granules are all commercially used in air-drying applications. Figure 16-6 shows how one of these processes can be adapted to regeneration with solar heat. A combination of dehumidification, air-to-air heat supply from solar and fuel can provide air-conditioning. This and other cycles, employing solar heated water and solar heated air for regeneration, have been experimentally investigated, but no commercial units are available.

REFRIGERATION COOLING WITH ROCK BED OR WATER STORAGE

Another cooling process that is not powered by solar energy, but is aided by the solar storage normally used for space heating, employs conventional cooling machines. A heat pump or vapor-compression

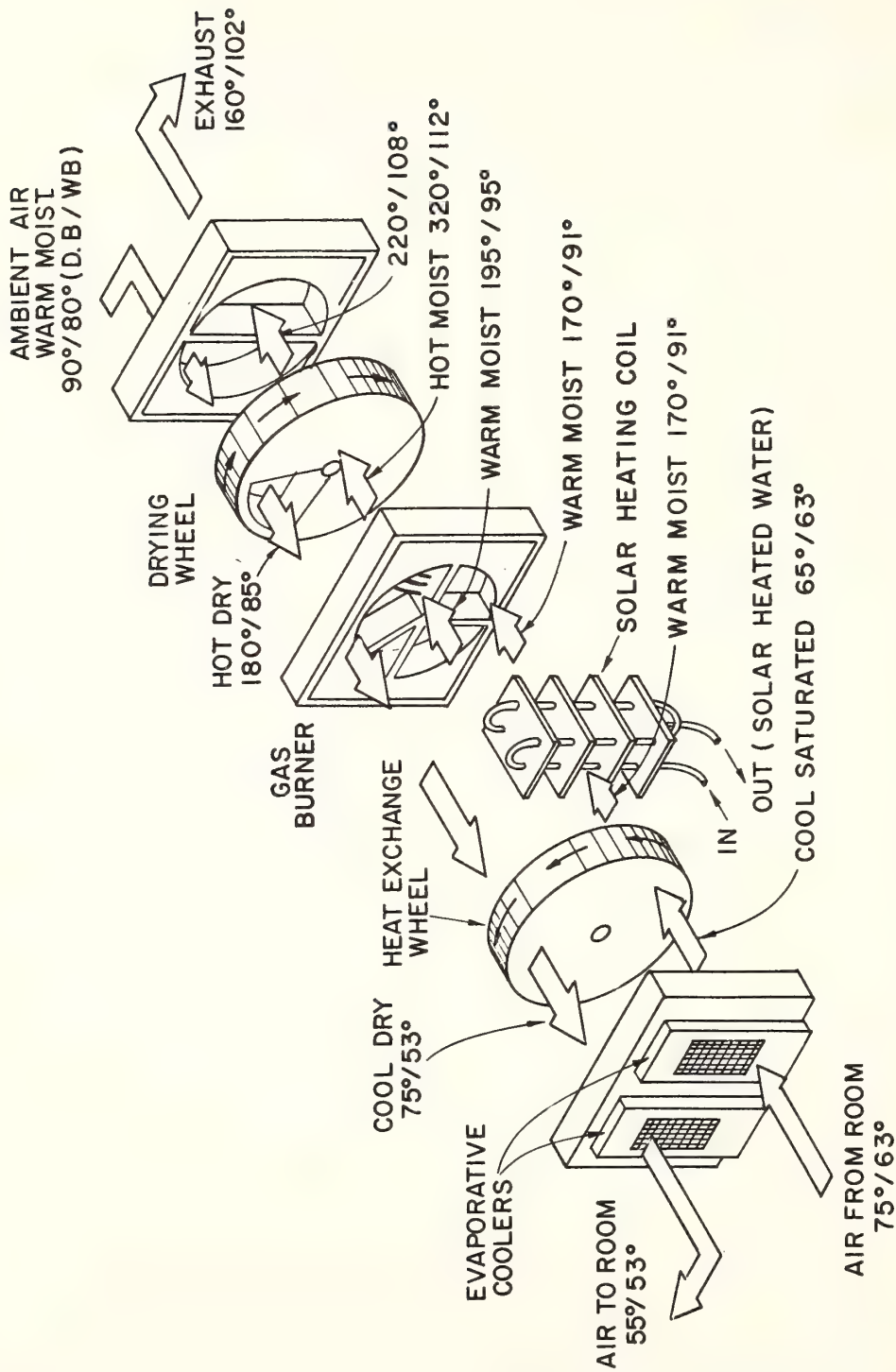


Figure 16-6. Solid Desiccant Dehumidification and Cooling System ("MEC" System)

air-conditioner supplies cool air to the storage unit during periods of low electric demand. Off-peak pricing permits lower cost air-conditioning by this night-time cooling of storage which then supplies cool air to the building by heat exchange during the daylight hours when power rates are usually higher. Water tanks may be used, with suitable heat exchangers, or pebble beds can be operated as direct air coolers. The possibility of moisture condensation and odor generation in a pebble bed used as cold storage must, however, be considered and avoided.

REFERENCES

1. American Society of Heating, Refrigeration, and Air Conditioning Engineers, ASHRAE Handbook of Fundamentals, ASHRAE, Inc., New York, 1972.
2. Barber, R.E., "Solar Air Conditioning Systems Using Rankine Power Cycles - Design and Test Results of Prototype Three-Ton Unit", Proceedings, Institute of Environmental Sciences Annual Meeting, Los Angeles, California, April 1975.
3. Bliss, R.W., Jr., "The Performance of an Experimental System Using Proceedings, United Nations Conference on New Sources of Energy, Volume 5, pp. 148-158, Rome, 1964.
4. Chung, R., Duffie, J.A., and Löf, G.O.G., "A Study of a Solar Air Conditioner", Mechanical Engineering, Volume 85, Number 8, pp. 31-35, August 1963.
5. Close, D.J., et. al., "Design and Performance of a Thermal Storage Air Conditioning System", Institute of Engineering Australia - Mechanical and Chemical Engineering Transcriptions, Volume MC 4, Number 1, pp. 45-54, May 1968.
6. Duffie, J.A. and Beckman, W.A., Solar Energy Thermal Processes, J. Wiley and Sons, Inc., New York, 1974.
7. Hay, H.R., and Yellott, J.I., "A Naturally Air-Conditioned Building", Mechanical Engineering, Volume 92, pp. 19-25, January 1970.
8. Johnston, R.C.R., "Regenerative Evaporative Air Conditioning with Dehumidification", AIRAH Transactions, Volume 21, Number 2, pp. 29-34. February 1967.
9. Threlkeld, J.L., Thermal Environmental Engineering, Second Edition, Prentice-Hall, Inc., New Jersey, 1970.
10. Ward, D.S., Weiss, T.A., and Löf, G.O.G., "Preliminary Performance of CSU Solar House I Heating and Cooling System", International Solar Energy Society Congress and Exposition, Los Angeles, California, July 1975.
11. Wilbur, P.J. and Mitchell, C.E., "Solar Absorption Air Conditioning Alternatives," Solar Energy, Vol. 17, pp. 193-199, 1975.

TRAINING COURSE IN
THE PRACTICAL ASPECTS OF
DESIGN OF SOLAR HEATING AND COOLING SYSTEMS
FOR
RESIDENTIAL BUILDINGS

MODULE 17

FUTURE PROSPECTS FOR SOLAR HEATING
AND COOLING SYSTEMS

SOLAR ENERGY APPLICATIONS LABORATORY
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INTRODUCTION

The solar systems described in this manual are examples of thousands that have been installed in the last three years. Performance data, mainly from well-instrumented systems, indicate that they can function satisfactorily in residential buildings. Air and liquids heated by solar energy in flat-plate collectors provide temperatures suitable for space heating, hot water supply, and cooling by absorption refrigeration. Although heat delivery efficiencies of the various systems differ between them, and fluctuate with climatic conditions, they should average about 30 percent of the solar radiation received. If this average can be improved by better components at equal or lower cost, the improvements are worthwhile.

A number of new features and components of systems are being investigated, and several might provide substantial improvement in system performance. Flat-plate collectors can be improved with selective coatings or redesigned to produce higher heat collection efficiencies. Evacuated collectors are inherently more efficient than conventional types, and developments which could reduce their cost can have important effects on commercial use. Heat storage in phase-change materials could reduce the space required for the solar heating system, and heat collection and storage by use of two immiscible liquids may result in system simplification and improved performance. If cooling equipment using solar-heated air can be developed, air-heating solar systems could be used throughout the year for heating and cooling. These and other developments may become important in the solar heating and cooling of buildings.

OBJECTIVE

This module contains descriptions of new concepts and developments in solar heating and cooling systems that could improve overall performance and economy. Its objective is to provide the trainee a basis for anticipating future developments and improvements in the systems described in this course and to recognize the research and development effort being devoted to component hardware in solar heating and cooling systems.

SOLAR COLLECTORS

The component which has the greatest influence on system performance and cost is the solar collector. Improvements in the collector which will increase its efficiency and reduce its cost can be particularly important. Among numerous possibilities are durable selective surfaces on absorbers, transparent honeycomb structures between absorbers and glazings, evacuated spaces between absorber surfaces and the glazings, and lower cost materials and assembly methods.

SELECTIVE SURFACES

Selective surfaces have high absorptance for solar radiation and low emittance for long-wave radiation. Substitution of such materials for non-selective black paints commonly used can increase solar collection more than the cost difference, thereby improving cost effectiveness of the collectors. Although already used in numerous commercially produced collectors, further improvements in optical properties and

durability, and reduction in cost, should be possible. Several selective coatings such as copper oxide on copper and black nickel on galvanized steel have been extensively used on solar water heaters in Australia and Israel, but performance and durability have not been completely satisfactory. Black chrome (electro-deposited chromium oxide on nickel-plated steel or copper) is the most widely used selective coating in the United States. Cost reduction could have an important influence on the future market for solar heating systems. It can be expected that virtually all commercial collectors for space heating and hot water supply will be provided with selective absorber coatings in the near future.

The radiation properties of several selective surfaces are listed in Table 17-1.

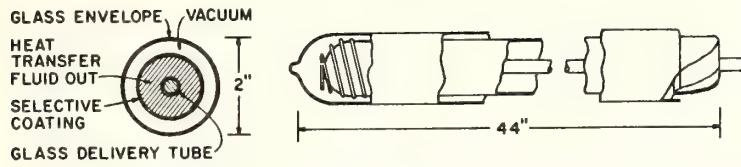
Table 17-1
Selective Surfaces Characteristics

Coating	Absorptance for Solar Radiation	Emittance for Thermal Radiation at 200°F
Chemically Oxidized Copper	0.90	0.15
Converted Zinc	0.90	0.071
Black Nickel	0.88	0.066
Black Chrome	0.92	0.085
Black Anodized Aluminum	0.93	0.07
Chemically Oxidized Stainless Steel	0.95	0.10

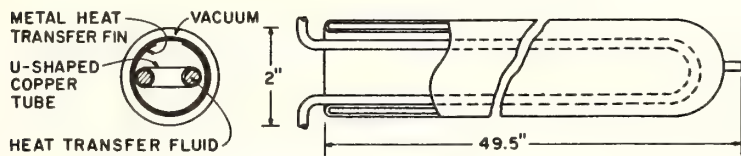
EVACUATED TUBE COLLECTORS

Removal of nearly all of the air in the space between the absorber and the glazing results in greatly reduced heat loss and a substantial efficiency improvement in solar collectors. There are a number of different designs that are being assembled and tested, and at least three manufacturers produce them in moderate quantities. Evacuated collectors produce more useful heat than equal areas of standard flat-plate collectors under the same sun and weather conditions, because heat conduction and convection through the evacuated space are negligible and, if the absorber coating is a selective surface, the radiation loss is small. Most flat-plate collectors have high efficiency with low inlet fluid temperatures, but have low efficiencies when the fluid temperature is near 200°F. The evacuated tube collector has a significant advantage when producing high temperature heat and can be used effectively with solar cooling units which require hot water at temperatures near 200°F.

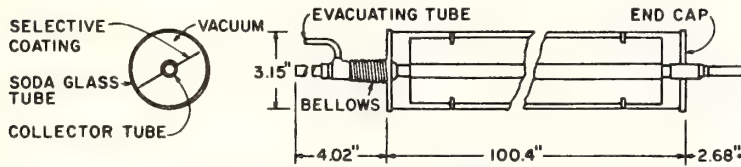
Several types of evacuated tube collectors are shown in Figure 17-1. Types A, B, and C are being produced on a semi-commercial scale and are being used in numerous demonstration heating and cooling installations, primarily in the United States and Japan. Types D and E are experimental evacuated tube collectors which have been tested in a few systems. In each design, a selective absorber surface is in an evacuated space. Types A, B, and E involve cylindrical-shaped absorber surfaces deposited on interior glass surfaces, while flat metal absorber surfaces are used in Types C and D. Double-walled glass tubes (Dewar flasks) are used in Types A and B, single tubes are used in C and D, and



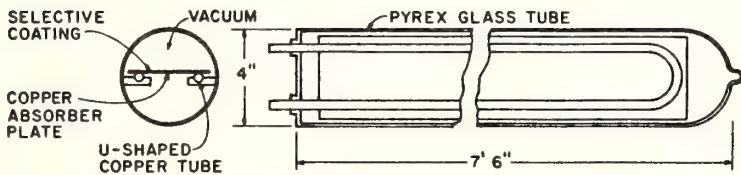
(a) Owens-Illinois Sunpak™ Double-Walled Evacuated Collector Tube.



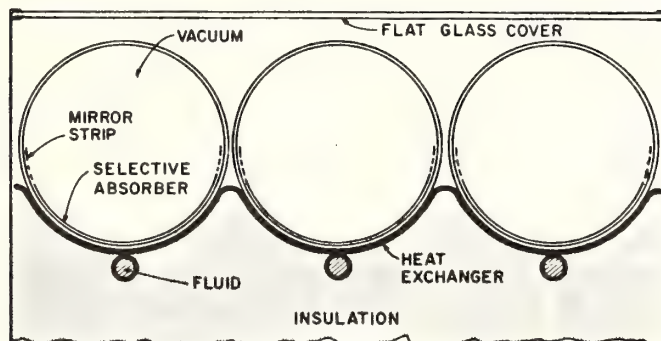
(b) General Electric TC-100 Solartron™ Double-Walled Evacuated Collector Tube.



(c) Sanyo Single-Walled Evacuated Collector Tube with Unidirectional Flow.



(d) Corning Single-Walled Evacuated Collector Tube.



(e) Philips Mark IV Evacuated Collector Tube (Section) with Flat Glass Cover.

Figure 17-1. Types of Evacuated Tube Collectors

single tubes with flat glass covers are used in E. In all designs except C, the fluid enters and leaves the tube at the same end, and in all designs except A, the fluid conduits are metal tubes (usually copper). In type A, the fluid is in contact only with glass, and in type E, the fluid flows in passages outside the glass tubes. Types A, B, and E, with minor modifications, may be used for air heating as well as water heating.

Collection efficiencies, based on solar radiation received by the sloping surface within the perimeter of the entire collector array, including space between tubes, are typically about 70 percent at temperatures for space heating (120°F to 150°F) and about 55 to 60 percent at the higher (200°F) temperatures required by lithium chloride absorption chillers. These efficiencies are substantially higher than good flat-plate collectors, particularly for cooling. Whether the superior efficiency justifies their higher current cost is doubtful, but the possibilities for manufacturing economies by automated production in large volume sustain commercial interest in this technology.

TRANSPARENT HONEYCOMBS

Heat loss from a flat-plate collector can be reduced by subdividing the air space between the absorber plate and the cover glass into small cells by use of a transparent "egg crate" or "honeycomb" structure. Convection movement of air in these small (one-quarter-inch or less) cells is substantially reduced, and thermal radiation from the absorber to the cover is also partially intercepted and suppressed. The transparent walls of these cells, however, permit the passage of solar

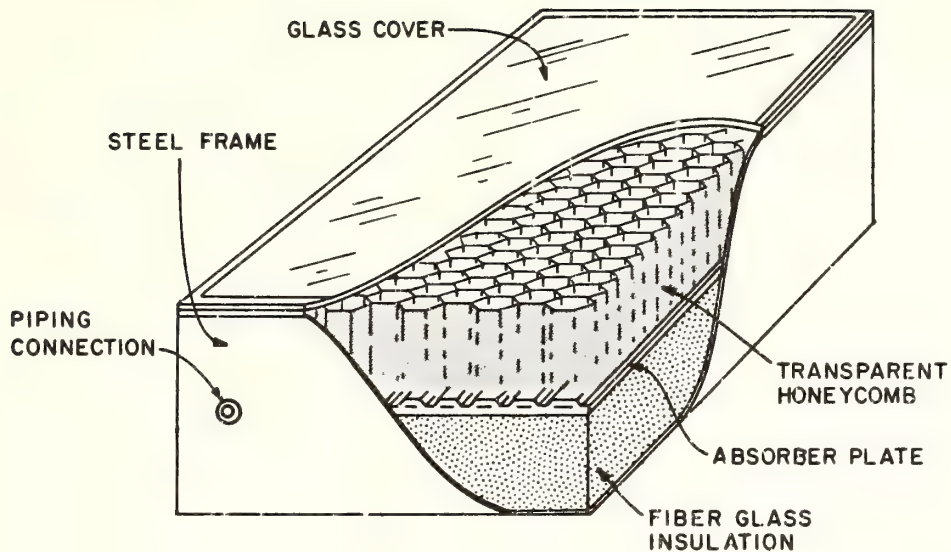


Figure 17-2. Transparent Honeycomb Collector

radiation with very little loss, even when obliquely intercepting the rays of the sun. The result of this design addition is increased solar collection efficiency. The principal problem in the use of this concept is achievement of sufficient durability of an economical material. Most transparent plastics cannot withstand the temperatures occasionally encountered when the fluid flow through the absorber is interrupted, melting or other type of failure then being probable. Thin glass honeycombs, as for example by use of a panel of side-by-side thin-walled glass tube sections, have been too costly for practical application. The future prospects for this concept appear to rest on the possibility for developing economical plastic materials and honeycomb structures

capable of withstanding the temperatures encountered in efficient solar collectors. If such materials are developed, honeycomb flat-plate collectors appear to have good prospects for practical use.

CONCENTRATING COLLECTORS

Concentrating collectors are being experimentally used when very high temperature fluid is needed to drive heat engines or for industrial process heat. If concentrating collectors can be designed to be less costly or more efficient than flat-plate collectors, to operate reliably, and to require little maintenance, such collectors also have potential use in space heating systems. Experience thus far has indicated otherwise, but there is considerable research underway and new designs for concentrating collectors are being developed. There has been experimental use of a sun-tracking linear focusing collector with a plastic Fresnel lens, and arrays of evacuated tubes with trough-shaped reflectors filling the spaces between tubes have been advertised for sale. Their costs are not competitive with flat-plate collectors in space heating applications nor with evacuated tube collectors (without concentration) for cooling applications.

THERMAL STORAGE

Future developments in the selection and application of materials in which heat can be stored by melting or other phase-change process may result in reduction in heat storage volume requirements. Hydrated

salts such as sodium sulphate decahydrate (Glauber's salt) and hydrated calcium chloride have been experimentally used, eutectic mixtures of two or more salts (having single melting points) have also been investigated, and several types of waxes have been evaluated. Table 17-2 contains a list of materials which have been used as phase-change heat storage media.

The potential advantage of this type of heat storage material is the smaller weight and volume required, compared with water and pebbles, for storage of an equal amount of energy. A second advantage, in some systems, is the storage of most of the energy at a constant temperature, i.e., the melting point of the material. Where volume limitations are very severe, phase-change thermal storage may become economically practical.

There are a number of disadvantages, however, in the use of these heat storage media. The need for transferring heat into and out of the solid and liquid compounds usually requires their containment in small elements, an inch or two in cross-section. Space between the elements must be provided for circulation of air or water from the solar collector and from the space being heated. Container cost and durability and the added volume of the free space in the storage bin or tank impose design and cost limitations.

A second disadvantage in most of the phase-change materials developed for solar space heating is their relatively low melting temperature. Although higher collector efficiency is obtained at low temperature, most space heating systems cannot advantageously use water

Table 17-2

Properties of Phase-Change Heat Storage Materials

Material	Formula	Melting Temperature °F	Heat of Fusion, Btu/lb	Liquid Density lbs/ft ³	Melting Characteristics
Glauber's Salt	$\text{Na}_2\text{SO}_4 \cdot 10\text{H}_2\text{O}$	88-90	108	91	Incongruent ⁽¹⁾
Disodium Phosphate	$\text{Na}_2\text{HPO}_4 \cdot 12\text{H}_2\text{O}$	97	114	95	Semicongruent
Sodium Thiosulfate	$\text{Na}_2\text{S}_2\text{O}_3 \cdot 5\text{H}_2\text{O}$	118-120	90	104	Semicongruent
Calcium Chloride	$\text{CaCl}_2 \cdot 6\text{H}_2\text{O}$	84-102	75	105	Semicongruent ⁽²⁾
Paraffin Wax	-	104-145	84	48-52	Congruent ⁽³⁾

- (1) Extensive separation of phase on repeated cycling.
 (2) Nearly congruent melting, suitable for repeated cycling.
 (3) Stability and flammability questions.

or air at the low temperature levels of 90°F to 100°F. Considerably larger heat exchange surfaces or much higher air circulation rates in warm air systems would have to be used.

Some of the phase-change materials, particularly the hydrated salts, on repeated melting and solidifying may permanently separate into liquid and solid portions, thereby defeating the phase-change heat storage capability. And in air systems, the relatively constant temperature of the heat stored is a disadvantage rather than an advantage, because the highly beneficial temperature stratification such as that occurring in pebble beds cannot be achieved.

The overriding consideration in heat storage selection is cost, and the presently suggested materials all are much more expensive, in assembled heat storage units, than tanks of water or bins of rocks. For storing half a million Btu, installed costs of advertised phase-change systems are two to five times as great as systems for sensible heat storage in rock and water.

Future prospects for practical use of phase-change storage in space heating systems depend on development of improved materials having considerably lower cost, longer life, and transitions in the 120° to 160° temperature range. If such materials can be identified and developed, their application could enhance the capability and economy in space heating systems.

An even more speculative possibility is the storage of solar heat in chemical reactions which absorb energy in the reaction process. By slight alteration in conditions such as the lowering of temperature or pressure, the reaction will reverse so that the stored energy is liberated as heat at a temperature near that at which solar collection

previously took place. Although such chemical processes are known, none is now a likely candidate for this application. Of particular advantage, if such a system can be developed, is the storage of the energy in materials which can be at room temperature. No heat losses or insulated containers would need to be involved. The possibilities for successful development of some such system do not appear promising at this time, but new discoveries might alter this situation.

HEAT EXCHANGER

A disadvantage in most liquid-heating solar systems is the temperature difference required for heat transfer to storage by means of a heat exchanger. The collector must operate at a temperature 10°F to 20°F higher than if no heat exchanger were used, and heat collection efficiency is therefore adversely affected.

Now under investigation is a heat exchanger-storage combination unit in which heat is transferred from liquid droplets that transport heat from the collector to water in the storage tank. A liquid that is immiscible with water is pumped through the solar collector and through the storage tank as droplets. If the density of the liquid is substantially different from that of water, the droplets will either rise or descend through the water in the storage tank. A schematic diagram of a heat exchanger-storage unit is shown in Figure 17-3. In this design, the collector liquid is heavier than water. It is delivered to the top of the tank, broken up into droplets at the perforated plate, and collected in the bottom cone. Because of the very large total surface areas of the droplets, the temperature difference between

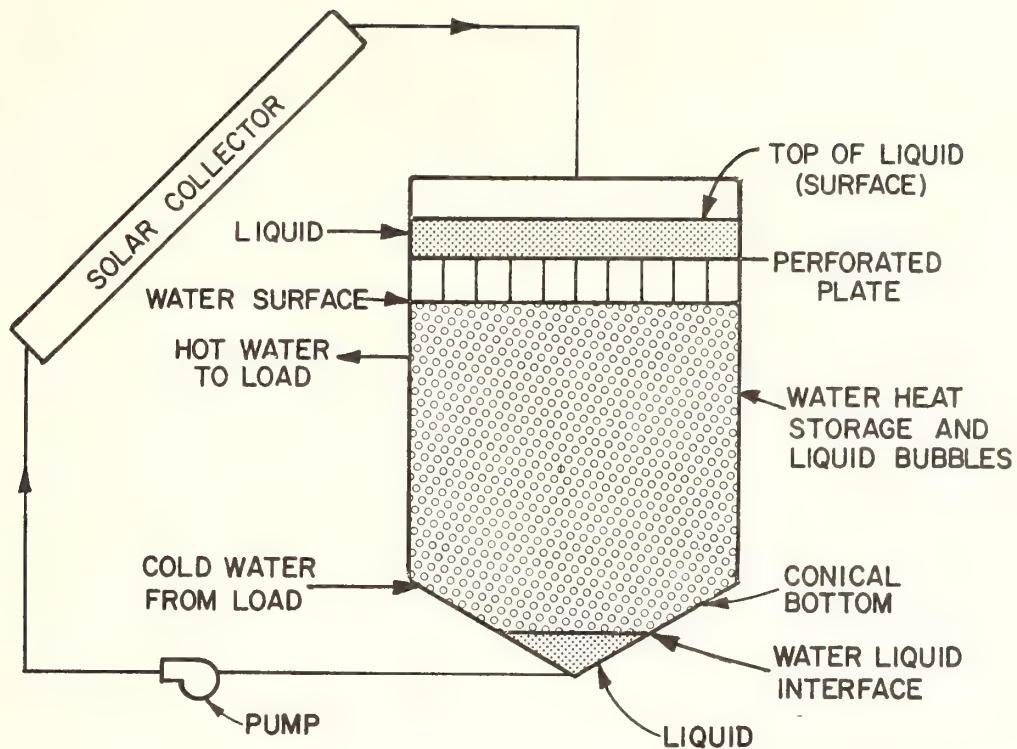


Figure 17-3. Direct Contact, Liquid-Liquid Heat Exchanger

them and the storage water is only about 1°F. Several liquids are candidates for this service, but of those listed in Table 17-3, diethyl phthalate is the only one that has received full-scale testing. Assessment of the future prospects for direct contact, liquid-liquid heat exchange-heat storage use depends on the results of further development work and economic evaluation.

SYSTEMS

At present the only commercially available small (3-ton) cooling unit that is operable with solar energy is a lithium-bromide absorption chiller. As mentioned elsewhere in this manual, there are several other solar cooling concepts under development, including heat engine-driven

Table 17-3
Properties of Possible Collector Fluids

Fluid	Freezing Point (°F)	Boiling Point (°F)	Specific Gravity at 200°F	Specific Heat	Cost (\$/gal)
Butyl benzy1 phthalate	-31	698	1.116	0.37	2.98
Cresyl diphenyl phosphate	-36	734	1.208	-	6.91
Dibutyl phthalate	-31	644	1.048	-	3.32
Diethyl phthalate	-41	568	1.120	0.38	3.46
Ethyl benzoate	-27	415	1.043	-	10.45
Tri o-cresyl phosphate	-13	770	1.162	-	8.63
Di (2 ethylhexyl) adipate	-76	782	0.927	-	3.79
Di (2 ethylhexyl) sebacate	-67	493	0.913	-	9.80
Therminol 55 [®]	-20	410	0.839	0.52	-
Dowtherm J [®]	-100	358	0.807	0.49	-

vapor-compression machines, absorption chillers with high temperature heat supply from solar concentrators, and desiccant cycles with solar regeneration of the drying agent. Significant effort is also being made in the development of so-called total energy systems, in which high temperature heat from solar energy is used to generate electricity and the low temperature "waste" heat is used to heat and cool a cluster of buildings. Such systems, if economically successful, would probably best be used in grouped facilities such as military bases but, with some variation, might serve a number of homes or apartment complexes.

In the long term, development of photovoltaic systems for electricity supply to residential buildings is a possibility. Solar electricity could then operate heating and cooling systems as well as other electrical equipment in the building. Whether the cost of photovoltaic systems will ever be low enough to be competitive with electricity generated from fossil or nuclear fuels is an open question, but a considerable amount of effort is being devoted to improve efficiency and reduce the costs.

Other system improvements which should increase the use of solar energy in buildings are hybrid designs consisting of passive as well as active components. As quantitative information and operating results of passive design techniques become available, superior designs can be identified and their cost effectiveness measured. Greater use of direct heating of residential space with passive systems may then decrease size requirements of the active components and thereby reduce overall costs.

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